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The Gasoline Automobile

ITS DESIGN AND CONSTRUCTION

VOLUME I

The Gasoline Engine

By

P. M. HELDT

Member, Society of Automotive Engineers
Engineering Editor, Automotive Industries

SEVENTH EDITION

P. M. HELDT
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PREFACE

IN making previous revisions of this volume the author always was handicapped by the fact that the type matter was in the form of electrotype plates, so that when any new matter was to be inserted an equal amount of old material had to be eliminated at the same place or near to it. In making arrangements for the present edition it was therefore decided to discard the plates and to reset all matter carried over from the previous edition.

The chapters dealing with the theoretical phases of the subject are not greatly altered, for the basic principles on which the gasoline engine works are still exactly the same as in 1911, when this volume was first written; they have been added to, however, to make the theoretical discussion more nearly complete and to take account of the special properties of the newer forms of engine, such as the eight-cylinder vertical and the V-type eights with two-plane crankshaft and with angles of V other than 90°.

The chapters dealing with the different principal parts of the engine, with engine characteristics and engine tests have been entirely rewritten. Approximately three hundred of the four hundred and fifty odd illustrations are new, as are also all of the plates at the end of the volume, with the exception of the three of the Class B Army truck engine.

What may be regarded as the backbone of the book always has been a discussion of the principles underlying and the practices followed in the design of automotive type internal combustion engines. Some space always has been devoted to methods of manufacture, but in the earlier editions this phase of the subject was dealt with rather briefly. Competition has become very keen in the automobile industry, and in order to meet conditions imposed by the downward trend in prices, manufacturers have had to lower their costs of production. This has led to great developments in manufacturing processes and equipment. Today, in order that a new car or a new engine may prove a commercial success, it is as essential that it be produced by the most advanced and most economical methods as that it be designed right from the operating standpoint. Economical production is possible only if the engine is specially designed with that end in view, hence engine design and production planning must go hand in hand.

A chapter has been added on Diesel-type or heavy oil automotive engines. Although oil engines do not come under the head of gasoline engines, there will hardly be any objection to the inclusion of material somewhat outside the scope defined by the title. The heavy oil automotive engine is as yet only in its infancy, but a number of truck and tractor engines of this type have been developed abroad to the commercial stage, and there can be little doubt that this engine will play an important part in the heavier branches of automotive transportation within a few years. That the movement is being pioneered mainly in Europe is due to the fact that the volatile hydrocarbon fuels used with carburetor-type engines are much more expensive there than in this country, hence the incentive for such development is greater there.

The title of this volume has been changed from The Gasoline Motor to The Gasoline Engine, because this latter term has been standardized by the Nomenclature Division of the Standards Committee of the Society of Automotive Engineers and is coming into constantly wider use. A change in the sub-title of the whole set, from Design and Construction to Design and Production, was also considered, as the term "Construction" is somewhat suggestive of the earlier days of the industry when automobiles were built rather than manufactured. It was not carried out, however, because if it had been the title of the work would have appeared differently in some of the volumes of the set than in others, which would have been inconvenient.

In gathering data and securing illustrations for the present edition the author has enjoyed the whole-hearted support of many firms in the industry, for which acknowledgment is herewith made. Particular thanks are due to the publishers of *Automotive Industries* of Philadelphia and *The Automobile Engineer* of London for the loan of drawings from which some of the plates in the Plate Supplement and some of the text illustrations were made.

As in the past, the author will be obliged to his readers for pointing out any inaccuracies.

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CHAPTER I

The Working Media—Gasoline and Air

The process by which mechanical power is generated in gasoline engines from the chemical potential energy of the fuel is substantially as follows: Gasoline is mixed with about fifteen times its weight of atmospheric air, in which it is vaporized. This explosive mixture is introduced into the working cylinders of the engine, where it is ignited and burned to water vapor and carbonic acid gas. The gases thus formed, together with the nitrogen of the air (which does not take part in the combustion), are heated to a high temperature by the heat of combustion of the fuel. In this way their pressure is greatly increased and they act upon the pistons of the engine, imparting to them a rectilinear motion, which by means of the connecting rods and crankshaft is transformed into rotary motion. Gasoline and air therefore are the working media of gasoline engines.

Gasoline is a colorless, mobile and relatively light liquid which is produced from crude petroleum by fractional distillation. According to older works on the subject, gasoline is that portion of the crude petroleum which distills over at 140° to 158° Fahr. and which has a specific gravity of 0.636 to 0.700. The product distilling at between 158° and 338° is known as benzine or naphtha. At the latter temperature kerosene begins to distill over. In recent years, as the demand for gasoline has increased, the refiners have marketed heavier and heavier products. At the beginning of 1926 the distillation range of the motor fuels sold in various parts of the United States extended from about 100° to about 425° F., so that not only the whole of the naphtha fraction but also some of the kerosene was included.

Chemical Composition of Gasoline—Gasoline is a mixture of various hydrocarbon compounds, most of which belong to the paraffin series. The principal paraffin hydrocarbons contained in motor fuel are:

Pentane, C_5H_{12}	Octane, C_8H_{18}
Hexane, C_6H_{14}	Nonane, C_9H_{20}
Heptane, C_7H_{16}	Decane, $C_{10}H_{22}$

There are others besides those mentioned, both lighter and heavier, the general chemical formula being C_nH_{2n+2}.

Since gasoline is not a definite chemical compound but a mixture of several such compounds in varying proportions, its various properties cannot be given with any degree of accuracy. The properties of the constituent compounds are well known, however, and some of them are given in the following table.

Table I.—Properties of Paraffin Hydrocarbons

Name	Per Cent of Carbon	Per Cent of Hydrogen	Boiling Point—°Fahr.	Specific Gravity	Vapor Density
Pentane	83.34	16.66	98	0.628	2.49
Hexane	83.28	16.72	157	0.672	2.97
Heptane	84.00	16.00	209	0.697	3.46
Octane	84.20	15.80	256	0.728	3.94
Nonane	84.38	15.62	300	0.741	4.43
Decane	84.50	15.50	338	0.754	4.91

Density Measurements—It is customary to distinguish the different grades of gasoline by their density or specific gravity. The specific gravity of gasoline is the ratio of the weight of a unit volume of gasoline at 60° Fahr. to the weight of the same volume of water at 39° Fahr. The density is usually expressed in terms of the Baumé scale, and in order to enable the reader to readily convert Baumé degrees into specific gravities, and vice versa, a conversion table covering that portion of the Baumé scale which can possibly be of use in this connection is given on the next page.

The Baumé and specific-gravity scales are connected by the following two equations:

$$\text{Deg. Bé.} = \frac{140}{\text{Spec. Gr.}} - 130 \text{ at } 63.5° \text{ Fahr.}$$

$$\text{Spec. Gr.} = \frac{140}{130 + \text{Deg. Bé.}} \text{ at } 63.5° \text{ Fahr.}$$

Since the standard temperature for making density tests of gasoline is 60° Fahr., instead of 63.5°, 0.002 is added to the specific gravity corresponding to any degree Baumé as calculated by means of these equations.

Baumé—Specific Gravity Conversion Table

Baumé Degrees	Specific Gravity	Baumé Degrees	Specific Gravity
56	0.755	69	0.706
57	0.751	70	0.702
58	0.747	71	0.699
59	0.743	72	0.695
60	0.739	73	0.692
61	0.735	74	0.689
62	0.731	75	0.685
63	0.727	76	0.682
64	0.724	77	0.679
65	0.720	78	0.675
66	0.717	79	0.672
67	0.713	80	0.668
68	0.709		

The density of gasoline is determined by means of an hydrometer, and the proper temperature at which to make the test is 60° Fahr. The reason for this is that gasoline, like all other substances, expands when heated and contracts when cooled. Therefore, if it is tested at a temperature either much greater or much less than 60°, the reading obtained will be incorrect and a correction factor must be applied. The specific gravity will naturally be less at higher temperatures and greater at lower temperatures. If

d_t = density at $t°$ F.

d_{60} = density at 60° F.

$$d_t = d_{60}[1 - \alpha(t - 60)],$$

α being the coefficient of expansion. According to tests made by Dr. W. Watson,

$\alpha = 0.00086$ for pentane,
$\alpha = 0.00074$ for hexane,
$\alpha = 0.00055$ for heptane,
$\alpha = 0.0007$ for gasoline (average of nine samples of different density).

.0005 — Gasoline

These tests of Dr. Watson's were made in 1910. Since that time the average density of gasoline has increased a great deal, and as, obviously, the coefficient of heat expansion increases as the density decreases, the coefficient for the present standard gasoline would be considerably lower than the value given above.

Distillation Curve—A much better criterion of the quality of a gasoline than its specific gravity or Baumé reading is its distillation curve. This curve shows the temperatures at which successive tenths of the fuel are boiled off. Its minimum ordinate is the temperature shown by the thermometer when the first drop appears in the still, and the maximum ordinate is the temperature at which the flask containing the gasoline becomes dry. This temperature is known as the end point. Composite distillation curves of the gasolines sold in the United States on July 1, 1924, and Jan. 1, 1925, are given in Fig. 1.

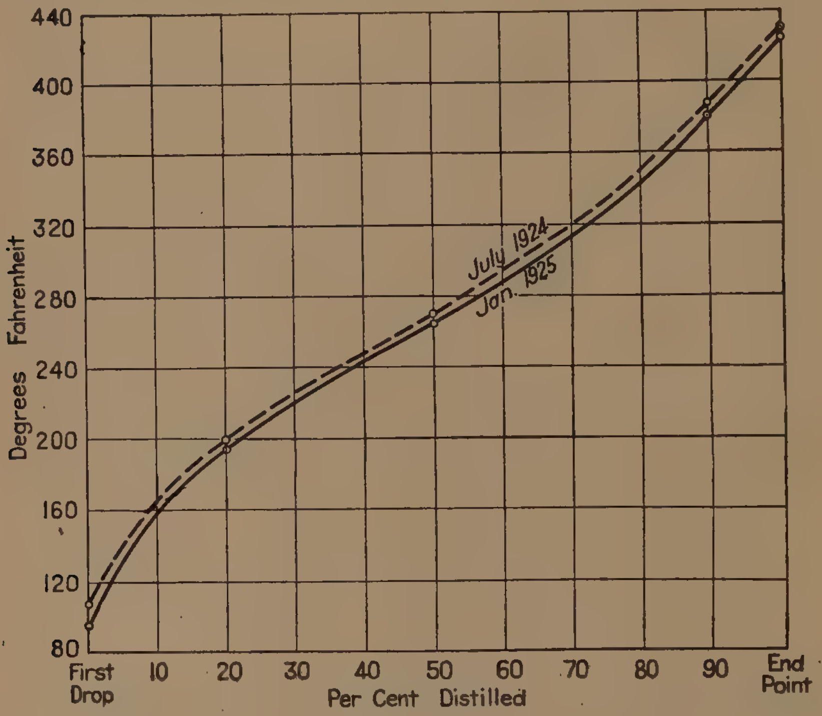


FIG. 1.—AVERAGE DISTILLATION CURVES OF AMERICAN GASOLINES.

Heat Value—The calorific value or heat value is the amount of heat liberated by gasoline when burned in an excess of air or oxygen. It is expressed in British thermal units per pound. One British thermal unit is the amount of heat required to raise one pound of water from 39° to 40° Fahr. The British thermal unit is equal to 778 foot-pounds.

There are two different heat values, the lower and the higher. The former is the amount of heat given out by a pound of gasoline when burned in an excess of air and when its products of combustion are cooled down to the boiling point of water, but the water vapor contained is not condensed. The higher heat value is the amount of heat given out when the gasoline is burned under the same conditions and the products of combustion are cooled down to 60° Fahr., so that the water vapor contained is condensed. Automobile engineers in the past have favored the lower heat value, because there is no possibility of practically utilizing the heat of condensation of the water vapor in the engine, but recently there has been a change in feeling in this matter, and many now favor the higher heat value, their argument being that the heat is present in the fuel, and whether or not it can be converted into mechanical work in any particular engine does not matter.

The calorific value of gasoline varies slightly with the composition, but for practical purposes sufficient accuracy will be insured by taking the lower heat value at 19,000 B.T.U. per pound and the higher heat value at 21,100 B.T.U. per pound. The lower heat values of the first three members of the paraffin series that are liquid at ordinary temperatures were found by Dr. W. Watson to be as follows: Pentane, 18,410 B.T.U.; hexane, 18,770 B.T.U.; heptane, 18,720 B.T.U. These values are somewhat lower than those found by other investigators. While the heat value of the different members of the paraffin series is practically the same on the weight basis, on the volume basis the heavier members of the series have an appreciably higher heat value.

Vapor Tension—The vapor tension corresponding to any temperature is the pressure under which liquid gasoline at that temperature will turn into a vapor when its surface is covered by a layer of its own vapor only. This pressure is counted from zero pressure and is equal to the gauge pressure plus 14.7 pounds per square inch. The vapor tension is equal to atmospheric pressure, or 14.7 pounds per square inch, at the boiling temperature of the liquid in the atmosphere. The vapor tensions of different components of gasoline at different temperatures are given below.

Vapor Tensions (Lbs. per Sq. In.)

Deg. Fahr.	50	100	150	200	250	300
Pentane	3.8	18.5				
Hexane	1.5	4	13			
Heptane	0.35	2.25	5.6	15		
Octane		0.9	2.7	6.4	13.5	
Nonane		0.4	1.35	3.5	7.7	14.8
Decane			0.7	2	4.7	9.7

Vapor tensions follow the following law, first established by Rankine:

$$\log t = C - \frac{C_1}{T} - \frac{C_2}{T^2},$$

where C, C_1 and C_2 are constants depending upon the particular liquid, t is the vapor tension in pounds per square inch and T the absolute temperature in degrees Fahr. Using the figures given in the table we get the following equation for the vapor tension of nonane:

$$\log t = 5.14 - \frac{2870}{T} - \frac{115{,}000}{T^2}.$$

The vapor tension of any composite liquid like gasoline depends, of course, almost entirely on its most volatile component.

Partial Pressure—Gasoline exposed to the atmosphere evaporates at any ordinary temperature, and the rate of evaporation depends upon the circulation of the air over its surface. As vapors are formed, they mix with the atmospheric air above the liquid and the gaseous pressure in the vicinity is then made up of the partial pressures of the air and gasoline vapor. The partial pressure of the gasoline vapor within any small cubic space is that pressure which would exist if the vapor alone occupied the space. Evaporation of the gasoline will continue until the partial pressure of its vapor in the layer of the atmosphere on its surface is equal to its vapor tension at the particular temperature. In still air the sum of the partial pressures is equal to atmospheric pressure, but when gasoline is used in an engine it is usually sprayed into the inlet tract, in which there exists a partial vacuum, so that for a given proportion of air and gasoline vapor the partial pressure of the latter is less, and this helps the vaporization.

Vapor Density—Gasoline vapor has a relatively high density, and when gasoline is spilled on a floor, the vapor rising from it will remain near the floor and spread rapidly to a considerable distance. The average vapor density of present-day gasoline is around 4.5; that is, at a given temperature and pressure a unit volume of gasoline vapor weighs 4.5 times as much as a unit volume of air. The vapor densities of the chief components of gasoline are given in Table I. The weight of a unit volume of atmospheric air at the standard temperature will be given later on, and from it the weight of gasoline vapor at different temperatures can be calculated.

Viscosity.—Tests to determine the viscosity of various grades of gasoline and the change of viscosity with temperature were made by J. S. G. Thomas and Dr. W. Watson, and their results were published in an Appendix to a paper by Dr. Watson on "Thermal and Combustion Efficiency of a Petrol Motor," presented to the Institution of Automobile Engineers (London). The following data are taken from that paper: The viscosity is expressed in C.G.S. units, but the figures are directly applicable to apparatus constructed according to English measures by means of the following formula: The weight of gasoline of density Δ which will flow in t seconds through a fine tube of diameter d inch and length l inches (l being great compared to d) when driven by a head H inches is

$$w = \frac{2.21\Delta^2 H d^4 t}{l\eta} \text{ pounds,}$$

where Δ = density or specific gravity of the gasoline;
H = head in inches;
d = diameter of the tube in fractions of an inch;
t = time in seconds;
l = length of the tube in inches,
and η = viscosity.

Viscosity of Gasoline

Density at 15°C.	Viscosity at— 5° C.	15° C.	25° C.
0.680*	0.00376	0.00342	0.00319
0.684	0.00380	0.00352	0.00332
0.704	0.00406	0.00380	0.00359
0.707	0.00445	0.00420	0.00398
0.714	0.00431	0.00404	0.00385
0.719	0.00445	0.00420	0.00398
0.720	0.00450	0.00421	0.00400
0.721	0.00454	0.00421	0.00400
0.725	0.00486	0.00454	0.00430

---

* Hexane.

It will be observed that with a few exceptions the viscosity increases uniformly with the density, and it in every case decreases uniformly as the temperature increases.

Latent and Specific Heats—Gasoline vaporizes readily at ordinary atmospheric pressures and temperatures, and it draws its heat of vaporization from the surrounding air. Little is therefore said, as a rule, regarding its latent heat or heat of vaporization, that is, the amount of heat required to transform one pound of liquid gasoline into a dry, saturated vapor from and at the boiling point. This factor varies from 160 B.T.U. for pentane to 114 for decane. The specific heat of liquid gasoline, that is, the amount of heat required to raise one pound one degree Fahr., is about 0.5 B.T.U.

Atmospheric Air—Dry atmospheric air is composed of 77 parts by weight of nitrogen and 23 parts by weight of oxygen. The air is capable of holding in suspension considerable amounts of water vapor, which have the effect of a diluent. In reality the atmosphere contains small quantities of other elements, but they are of no consequence in the operation of a gasoline engine and may be neglected. The oxygen of the air combines with the elements of the fuel to form new compounds, while the nitrogen does not take part in the combustion. At ordinary atmospheric pressure (29.92 inches of mercury) air at a temperature of 32° Fahr. weighs 0.0807 pound per cubic foot. Air expands 1/492 of its volume at 32° Fahr. for every degree Fahr. increase in temperature.

Ratio of Chemical Combination—We are now in position to make a calculation of the amount of air required to burn a certain quantity of gasoline. For the sake of simplicity we will assume that this gasoline consists solely of nonane (C_9H_{20}). This assumption is justified because nonane is intermediate in its composition and properties between the lightest and heaviest constituents of commercial motor gasoline, and, besides, the proportion of carbon to hydrogen does not differ much in the different members of the paraffin series found in gasoline. Carbon has an atomic weight of 12 and hydrogen of 1, hence the proportions by weight of the two elements in nonane are

$$9 \times 12 = 108 \text{ parts of carbon to}$$

$$20 \times 1 = 20 \text{ parts of hydrogen,}$$

or, expressing the proportions on a percentage basis,

84.38 per cent of carbon,
15.62 per cent of hydrogen.

When there is plenty of air mixed with the gasoline, on ignition the carbon of the fuel will combine with the oxygen of the air to form carbon dioxide gas (CO_2), and the hydrogen will combine with the oxygen to form water vapor (H_2O). Remembering that oxygen has an atomic weight of 16, we may write the reactions as follows:

$$12C + 32O = 44CO_2$$
$$2H + 16O = 18H_2O$$

The coefficients in each of these equations show the weight proportions in which the elements combine. Now let us take one pound of gasoline or nonane. We then have 0.844 lb. of carbon and 0.156 lb. of hydrogen, and by using the proportions given in the above equations we can find the quantities of oxygen required for the combustion of this carbon and hydrogen.

$$0.844C + (\tfrac{32}{12})0.844O = (\tfrac{44}{12})0.844CO_2,$$
$$0.156H + (\tfrac{16}{2})0.156O = (\tfrac{18}{2})0.156H_2O.$$

Carrying out the multiplications,

$$0.844C + 2.251O = 3.095CO_2$$
$$0.156H + 1.248O = 1.404H_2O$$

Total oxygen required 3.499

But it was stated above that this oxygen is mixed in the atmosphere with nitrogen in the proportion of 23 to 77; consequently, 3.499 lbs. of oxygen is mixed with $77/23 \times 3.499 = 11.714$ lbs. of nitrogen or is contained in $100/23 \times 3.499 = 15.2$ lbs. (approx.) of air. Therefore, one pound of gasoline (nonane), theoretically requires 15.2 pounds of air for its complete combustion. The process of combustion in the gasoline engine may therefore be written in chemical terms as follows:

$$\overbrace{0.844C + 0.156H}^{\text{1 pound gasoline}} + \overbrace{3.499O + 11.714N}^{\text{15.2 pounds air}}$$
$$= 3.095CO_2 + 1.404H_2O + 11.714N$$

Effect of Variations in Proportion—Since the mixture of gasoline vapor and air is never perfectly homogeneous, it is advisable to have a slight excess of air in the mixture if maximum economy is desired, and a slight excess of fuel if it is of more importance that the maximum possible power output be obtained from the engine. Tests have shown that the maximum economy is obtained from mixtures whose proportion ranges between 15:1 and 17:1, whereas the maximum power is obtained with a mixture of about 12.5 : 1. If insufficient air is present, the hydrogen will burn first, and then as much of the carbon as there is air for will burn to carbon dioxide. Some of this carbon dioxide will then combine with additional carbon to form carbon monoxide (CO). Combustion to carbon monoxide not only gives off a much smaller amount of heat per pound of fuel, but carbon monoxide is a very poisonous gas and makes a foul exhaust.

Difference between Higher and Lower Heat Values—Reference was made in the foregoing to the two methods in vogue for determining the heat values of hydrocarbon fuels. The theoretical difference between the two heat values can readily be calculated. We found that in burning one pound of gasoline 1.404 pounds of water vapor is formed. The latent heat of water is 965.7 B.T.U. per pound, and the cooling of the water from 212° to 60° gives a further 152 B.T.U. per pound, making a total of 1118 B.T.U. per pound. Of the other two constituents of the products of combustion, carbon dioxide and nitrogen, the former has a specific heat of 0.216 and the latter of 0.244. The total heat gained in condensing the water vapor and cooling all of the products of combustion of one pound of gasoline from 212° to 60° Fahr. is therefore

$$\begin{array}{lr} H_2O, & 1.404 \times 1118 = 1570 \\ CO_2, & 3.095 \times 152 \times 0.216 = 101 \\ N, & 11.714 \times 152 \times 0.244 = 434 \\ \hline \text{Total.................} & 2105 \text{ B.T.U.} \end{array}$$

Theoretical Heat Value of Gasoline—The total heat liberated in the combustion of gasoline is the sum of the heats liberated by the combustion of the carbon and the combustion of the hydrogen, minus the heat required to break up the hydrocarbon molecules. In the combustion of carbon to carbon dioxide the heat liberated per pound of carbon consumed is 14,650 B.T.U. In the combustion of hydrogen to water vapor the heat liberated per pound of hydrogen consumed is 52,920 B.T.U. The dissociation of nonane into hydrogen and carbon requires 1572 B.T.U. per pound. If now we take the quantities of carbon dioxide and water vapor produced by the combustion of one pound of nonane as found in the foregoing, we may write the following heat balance:

$$\begin{array}{rr} 0.8438 \times 14{,}650 = 12{,}355 & 20{,}615 \\ 0.1562 \times 52{,}920 = 8{,}260 & -\ 1{,}572 \\ \hline 20{,}615 & 19{,}043 \text{ B.T.U.} \end{array}$$

Gasoline mixtures containing the relative proportions of fuel and air required for complete combustion are readily ignited by an electric spark. When the proportion differs much from this standard either way it becomes more difficult to ignite the mixture and once ignited, the flame travels through it less rapidly. The limiting proportions for which ignition is still possible at atmospheric pressure are about 1:4 (rich mixture) and 1:20 (lean mixture). These figures refer to 73° Baumé gasoline at atmospheric temperature. At high temperatures the lean mixture limiting ratio is reduced somewhat and the rich mixture limiting ratio is increased.

Feul oil. — 20,000 Btu/# — Low
std.

CHAPTER II

The Laws of Gases—Thermodynamics

Heat is a form of energy and is convertible into other forms of energy, which in turn are convertible into it. Energy is defined as the capacity for performing mechanical work. Experiments conducted by Helmholz, Joule and Mayer during the past century proved that energy is indestructible; it can be converted from one form into another, but the total amount of it never changes. This does not imply, however, that the conversion is always complete. In converting heat energy into mechanical energy it is rare that more than 35 per cent of the heat energy appears as mechanical energy, and generally the efficiency is very much less. The rest of the heat is lost by transmission through the walls, etc.

The most familiar form of energy is mechanical energy, which is measured in foot-pounds. One foot-pound is the energy expended in raising one pound of matter one foot against the force of gravity. Heat energy is measured in British thermal units. The B. T. U., as already stated, is that amount of heat which is required to raise one pound of water from 39° to 40° Fahrenheit. Experiments showed that

$$1\ B.\ T.\ U. = 778\ \textit{foot-pounds},$$

and the figure 778 is known as Joule's equivalent or the mechanical equivalent of heat. It is represented in mathematical discussions by the letter J.

There are two broad principles underlying the conversion of heat into mechanical energy and the reciprocal process, known respectively as the First and the Second Law of Thermodynamics. They may be stated as follows:

First Law of Thermodynamics—When work is done by the expenditure of heat, the quantity of heat consumed is a measure of the quantity of work done, or of the energy acquired in a new form; and conversely, if work is converted into heat energy the conversion takes place in a definite ratio.

Second Law of Thermodynamics—If the total actual heat of a homogeneous and uniformly hot substance be conceived to be divided into any number of equal parts, the effects of these parts in causing work to be performed are equal (Rankine).

Perfect Gases—It has been found desirable in scientific discussions to distinguish between perfect gases—those far from their temperature of liquefaction—and imperfect gases or vapors—those near their point of liquefaction. Although gasoline vapor forms one of the constituents of the combustible charge used in gasoline engines, it is convenient in discussing the theory of these engines to consider the charge as a perfect gas.

When a quantity of a perfect gas is confined in a closed vessel it exerts a uniform pressure over the entire interior surface of the vessel. This pressure is known as the gas pressure and is usually expressed in pounds per square inch. If the volume occupied by the gas is changed, the pressure will generally be changed. It has been found that if the volume of the gas is reduced in a certain proportion—the temperature of the gas remaining the same—the pressure will be increased in the same proportion; inversely, if the volume is increased in any proportion, the temperature of the gas remaining the same, the pressure will be reduced in the same proportion. This law is known as Boyle's or Marriotte's law, and is mathematically expressed by the equation

$$PV = Constant,$$

where P is the pressure and V the volume. Compression of the gas not accompanied by any change in the temperature of the same is known as isothermal compression, and expansion under the same conditions as isothermal expansion.

Ordinarily, however, the compression and expansion of gases are accompanied by temperature variations, as will be explained farther on.

Absolute Temperature Scale—When a gas is heated, if it is free to expand, it will do so, and a certain relation has been found to exist between the volume increase and temperature increase. It is known that the ordinary thermometric scale is entirely arbitrary, the freezing point of water being called 32°, the boiling point of water 212° and the entire scale divided into parts equal to 1/180 of the distance between these two points. But if we fix our zero at what on the ordinary Fahrenheit scale would be —460° and measure temperatures from that point, then the increase in volume of a perfect gas (the pressure remaining constant) will be directly proportional to the increase in the temperature of the gas, or the increase in pressure (the volume remaining constant) will be directly proportional to the increase in temperature.

This point on the temperature scale, viz., —460° Fahr., is called the absolute zero, and temperatures measured from it are called absolute temperatures. The above relationship in its most general form is known as Charles' law and is expressed mathematically as follows:

$$\frac{PV}{T} = Constant,$$

P being the absolute pressure; V, the volume, and T the absolute temperature.

Specific Heats—The amount of heat required to raise the temperature of any substance by one degree Fahrenheit depends upon what is known as its specific heat. It has already been pointed out that it requires one British thermal unit to raise one pound of water from 39° to 40° Fahrenheit, and water is considered to have a specific heat of unity; hence the specific heat of other substances is equal to the amount of heat expressed in British thermal units required to raise the temperature of one pound of these substances one degree Fahrenheit. In the case of gases, however, it makes a difference whether the rise in temperature is accompanied by a rise in the pressure of the gas or by an increase in the volume. If the gas is heated at constant volume, the heat in British thermal units required to raise one pound 1 degree Fahrenheit is known as the specific heat at constant volume (C_v), while if the gas is heated at constant pressure the heat in thermal units required to increase the temperature 1 degree Fahrenheit is known as the specific heat at constant pressure $(C)_p$. The specific heat of air at constant volume has been found by experiment (Regnault) to be 0.1691. From this we can calculate the specific heat of air at constant pressure as follows:

Consider a cylinder of such a bore that its cross-section measures just one square foot. A piston rests in this cylinder with its plane head exactly one foot from the plane cylinder head. The space back of the piston then measures one cubic foot and we will consider this space filled with air at the normal atmospheric pressure corresponding to a barometric pressure of 29.92 inches of mercury column and a temperature of 32° Fahr., or 491.6° absolute. We will assume that the piston makes an air-tight joint with the cylinder wall but that it can move in the cylinder without friction.

Now let heat be applied to the air confined in the cylinder so that its temperature is raised. As the temperature rises the pressure of the air tends to rise, but we have assumed that the piston moves in the cylinder without friction; and, since its outer face is subjected only to atmospheric pressure, immediately the pressure of the confined air rises above atmospheric, the piston moves outward, thus keeping the pressure within down to atmospheric. Let us keep on supplying heat until the temperature of the air within the cylinder has been raised through one degree; then, according to Charles' law, the volume of the air will have increased in the proportion of 492.6/491.6 or by 1/491.6 cubic foot. In other words, the piston will have moved a distance of 1/491.6 ft. In making this motion it had to overcome the atmospheric pressure of 14.6967 lbs. per. sq. in., or 2116.3 lbs. per sq. ft. As the piston head area is equal to one square foot the resistance to be overcome is 2116.3 lbs., and as it moves a distance of 1/491.6 ft. against this resistance, the work done in "pushing back the atmosphere" is

$$2116.3/491.6 = 4.305 \text{ ft.-lbs.},$$

which is equal to

$$4.305/778 = 0.005533 \text{ B.T.U.}$$

Consequently, if a cubic foot of air is raised in temperature from 32° to 33° Fahr. under these conditions, in pushing back the atmosphere there must be expended an amount of energy equal to 0.005533 B.T.U. Now, one cubic foot of air under normal atmospheric pressure and at 32° Fahr. weighs 0.080728 lb. Hence the heat required per pound of air per degree Fahr. is

$$0.005533/0.080728 = 0.0686 \text{ B.T.U.}$$

Adding this to the energy required to raise the temperature of the air from 32° to 33° at constant volume, we find for the heat required to raise it through the same temperature gradient at constant pressure

$$0.1691 + 0.0686 = 0.2377 \text{ B.T.U.}$$

We therefore have

Specific heat of air at constant volume, $C_v = 0.1691$
Specific heat of air at constant pressure, $C_p = 0.2377$

The ratio of these two factors

$$0.2377/0.1691 = 1.405,$$

plays quite an important role in thermodynamics.

Consider again the aforesaid cylinder with a cross-section of 1 square foot, and containing a volume V of air at the temperature T and pressure P (both absolute). Now, let heat be applied to this gas until the temperature becomes T_1 and the volume V_1. Part of the heat is expended in raising the temperature of the air and the rest in pushing back the atmosphere or other pressure holding the piston in place; that is, in doing external work. We will consider the quantity of air in the cylinder a unit quantity and P the pressure per square foot. As regards units, we can, of course, make them anything we choose, only we must use the same units throughout the discussion. The foregoing are here chosen in order to eliminate coefficients which would simply complicate the work, but would not affect the result.

The heat required to raise the temperature of the gas in this operation is the same as would be required to raise the temperature the same amount at constant volume, viz.,

$$C_v. \ (T_1 - T).$$

The total heat required being $C_p(T_1 - T)$, it follows that the difference $(C_p - C_v)$ $(T_1 - T)$ is expended in doing external work. This work, however, can also readily be expressed in other terms. Since the cross-section of the cylinder is exactly 1 square foot, the volume increase in cubic feet is the same as the motion of the piston in feet, and *vice versa*. Therefore, when the volume increases from V to V_1 the piston moves $V_1 - V$ feet, and since it moves against the pressure P, the work done is

$$P(V_1 - V) \textit{ foot-pounds,}$$

which is equivalent to

$$\frac{P}{J}(V_1 - V) \textit{ heat-units,}$$

J being the mechanical equivalent of heat.

Equating the two expressions for the external work done we have

$$(C_p - C_v)(T_1 - T) = \frac{P}{J}(V_1 - V). \quad . \quad . \quad . \quad (1)$$

But since the pressure P remains constant, we have

$$\frac{V}{T} = \frac{V_1}{T_1} \text{ (Charles' law),}$$

or

$$V_1 = T_1\frac{V}{T}.$$

Substituting this value of V_1 in equation (1) we obtain:

$$(C_p - C_v)(T_1 - T) = \frac{P}{J}\left(T_1\frac{V}{T} - V\right).$$

Dividing both sides by

$$\left(\frac{T_1}{T} - 1\right).$$

we have

$$(C_p - C_v)T = \frac{P}{J}V,$$

which by transformation gives

$$\frac{PV}{T} = J(C_p - C_v).$$

But by Charles' law

$$\frac{PV}{T} = \textit{Constant},$$

consequently,

$$J(C_p - C_v) = \textit{Constant}.$$

This constant is generally designated by R, so that we may write:

$$\frac{PV}{T} = J(C_p - C_v) = R. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

For air,

$$R = 778(0.2377 - 0.1691) = 53.37,$$

hence

$$\frac{PV}{T} = 53.37.$$

We can also obtain the value of R by inserting values for the quantities of the right-hand side of equation (2). A cubic foot of air (the unit mass with which we have been dealing) at normal atmospheric pressure and 32° Fahr. or 491.6° abs. weighs 0.0807 lb. The pressure per square foot is

$$144 \times 14.6967 = 2116.3 \text{ lbs.}$$

Hence,

$$\frac{PV}{T} = \frac{2116.3}{491.6} \times \frac{1}{0.0807} = 53.36.$$

From the equation

$$\frac{PV}{T} = 53.37$$

we have

$$\frac{P}{TW} = 53.37,$$

and

$$W = \frac{P}{T \times 53.37},$$

where W is the weight of one cubic foot of air at a pressure of P pounds per square foot and an absolute temperature of T deg. If P is expressed in inches of mercury column and T in degrees F. we get

$$W = \frac{P(2116.3/29.92)}{(T + 459.6)53.37}$$

$$= \frac{1.3256P}{459.6 + T} \text{ lb. per cu. ft.}$$

This equation enables us to readily determine the weight of atmospheric air at any pressure and temperature.

It will be noted that in dealing with gases there are three main factors, viz., the volume, the temperature and the pressure. We have thus far always assumed one of these factors to be constant and then investigated the change of the second with a change in the third. But in practical work with gases all of the foregoing factors often change simultaneously.

Adiabatic Changes of State—According to the principle of the first law of thermodynamics, if heat is supplied to a quantity of air or gas the amount of heat thus supplied will be equivalent to the heat used for raising the temperature of the air or gas plus the energy expended in doing external work. Considering, therefore, a small amount of heat, H, supplied, which will cause an infinitesimal increase, dT, in the temperature of the air and an infinitesimal increase, dV, in the volume, we may write:

$$H = C_v dT + \frac{PdV}{J}.$$

From this it readily follows that if any changes take place in the state of the air without the total amount of heat in it being either added to or subtracted from ($H = 0$), then

$$C_v dT + \frac{PdV}{J} = 0.$$

Such changes in the state of gas, not accompanied by the addition of heat from outside, or the subtraction of heat from the gas, are known as adiabatic (equal heat) changes.

The limiting ratio of temperature to volume change is therefore

$$\frac{dT}{dV} = -\frac{P}{JC_v}. \qquad (3)$$

We found that

$$\frac{PV}{T} = R = J(C_p - C_v),$$

from which it follows that

$$PV = TR = TJ(C_p - C_v).$$

Differentiating with respect to P, V and T, we have:

$$PdV + VdP = J(C_p - C_v)dT;$$

$$J\frac{dT}{dV}(C_p - C_v) = P + V\frac{dP}{dV}.$$

Substituting the value of $\frac{dT}{dV}$ found in (3), we have

$$-P\left(\frac{C_p}{C_v}\right) + P = P + V\frac{dP}{dV},$$

which can be changed to

$$\frac{dP}{P} = -\left(\frac{C_p}{C_v}\right)\frac{dV}{V}.$$

Now, it may be found in text books on the Integral Calculus that the integral of a differential expression of the form dx/x is $\log x$. Hence, integrating, we have:

$$\log P = -\frac{C_p}{C_v}\log V + Constant.$$

Transposing,

$$\log P + \frac{C_p}{C_v}\log V = Constant,$$

and taking antilogs,

$$PV^{\frac{C_p}{C_v}} = Constant.$$

The ratio of the specific heats C^p/C^v is generally designated by γ, so we may write:

$$PV^\gamma = Constant. \quad \ldots\ldots \quad (4)$$

This gives us an expression connecting the pressure and volume for adiabatic changes in the state of the gas. The temperature is not a factor in this equation, but an expression for it may readily be found as follows:

From the equation,

$$\frac{PV}{T} = R,$$

we find that

$$P = \frac{TR}{V},$$

and multiplying both sides by V^γ,

$$PV^\gamma = \frac{TRV^\gamma}{V} = Constant,$$

or, since R is a constant

$$TV^{\gamma-1} = Constant. \quad \ldots\ldots \quad (5)$$

We also may write the foregoing equation in the form

$$V = \frac{TR}{P},$$

so that

$$V^\gamma = \left(\frac{TR}{P}\right)^\gamma;$$

and multiplying both sides by P.

$$PV^\gamma = P\left(\frac{TR}{P}\right)^\gamma = Constant,$$

from which it follows that

$$\frac{T^\gamma}{P^{\gamma-1}} = Constant. \quad \ldots\ldots \quad (6)$$

Summarizing, for adiabatic expansion or compression of gas the following three equations hold:

$$PV^\gamma = Constant;$$

$$TV^{\gamma-1} = Constant;$$

$$\frac{T^\gamma}{P^{\gamma-1}} = Constant.$$

The following table gives the values of C_p, C_v and γ for the different gases involved in gasoline motor work:

Gas	C_p	C_v	γ
Air	0.2377	0.1691	1.405
Oxygen	0.2175	0.1551	1.370
Nitrogen	0.2438	0.1727	1.412
Hydrogen	3.409	2.4110	1.414
Carbon monoxide	0.2450	0.1736	1.411
Carbon dioxide	0.2869	0.172	1.668
Steam	0.4805	0.370	1.298

Temperature of Gasoline Combustion—We found that one pound of gasoline in burning evolves approximately 19,000 B.T.U. This heat energy in a gasoline engine as now commonly used is applied to the gas at a constant volume, and all of it therefore goes to raise the temperature of the gas. The three constituents of the gases in the cylinder after the charge has been burned are nitrogen, carbon dioxide and water vapor, and considering the proportions in which they are present and the specific heat at constant volume of each, we should expect the specific heat of the products as a whole to be about 0.20. Assuming that one part of gasoline vapor is mixed with 15 parts of air, we should expect a rise in temperature on combustion of

$$\frac{19{,}000}{16 \times 0.20} = 5900^\circ \mathit{Fahr.}\ (\mathit{appr.}).$$

In reality the temperature never exceeds 3500° Fahrenheit. Various reasons have been advanced in explanation of this discrepancy between theory and practice. The first and most plausible explanation is that the specific heat of gases, instead of being constant, increases with the temperature. This subject has been experimentally investigated by Mallard and Le Chatelier. From the results of their experiments it is found that there is a linear relation between the specific heat at constant volume and the temperature of the gas, and that the specific heats may be represented by the following equations:

$$\begin{aligned}
&\text{For } CO_2, && C_v = 0.1408 + 0.0000463T;\\
&\text{For } H_2O, && C_v = 0.279 + 0.0001011T;\\
&\text{For } N_2, && C_v = 0.171 + 0.0000120T;\\
&\text{For } O_2, && C_v = 0.150 + 0.0000104T.
\end{aligned}$$

In these T is the regular Fahrenheit temperature and C_p the average specific heat up to the temperature T. It will be noted that the temperature coefficient for those gases which have the lowest temperature of liquefaction (N_2 and O_2) are the lowest; in other words, for these gases the specific heat is most nearly constant. If we determine the temperature of combustion by making use of the foregoing specific heats, the result is still considerably above observed results, and the increase in the specific heats at high temperature, therefore, does not fully explain the discrepancy between calculated and observed results.

Other explanations are that the cooling effect of the walls prevents the gas from attaining the calculated temperature, that compounds formed by the combustion, viz., carbonic acid gas and water vapor, become dissociated at high temperatures, whereby heat energy is absorbed, and that the chemical reactions are not yet terminated at the moment the maximum temperature is reached, so that this temperature does not depend upon the total heat of combustion of all of the gas. This latter theory is referred to as that of ''after burning.'' The fact that the pressure and temperature of explosion are higher in an engine with hemispherical combustion chamber than in an engine with valve pockets on opposite sides proves that the cooling influence of the walls and ''after burning'' affect the temperature of explosion.

CHAPTER III

The Otto Cycle

The great majority of automobile engines are operated on what is known as the Otto or four-stroke cycle. By a cycle is meant the succession of operations in a cylinder with a single charge of explosive mixture. The Otto or four-stroke cycle comprises the following four operations, succeeding one another in the order given:

Admission of the charge to the cylinder.
Compression of the charge.
Combustion of the charge (which includes its ignition and expansion).
Exhaustion of the products of combustion.

Each of these four operations occupies the time of one piston stroke. This cycle was originally proposed by Beau de Rochas, a Frenchman, who in 1862 took out a French patent on an engine employing it. Its first practical application, however, is due to Dr. N. A. Otto, a German engineer, who in 1876 patented, and in 1878 exhibited at the World's Fair in Paris the first four-stroke cycle engine of the general type in common use today.

Admission—The first stroke of the Otto cycle is known as the admission or intake stroke (Fig. 2). The piston moves outward in the cylinder, thereby creating a rarefaction of the air or gas in the outer or working end of the cylinder (generally known as the combustion chamber). At some point in the wall of the combustion chamber is located a valve which, at the proper time, is opened to place the chamber in communication with the source of combustible mixture, the carburetor. This valve begins to open a little after the beginning of the admission stroke and closes a little after the admission stroke has been completed.

When the engine is rotating at low speed the degree of vacuum or suction in the cylinder during the admission stroke will be very small, and it will, under any conditions, be the smaller the greater the cross section of the inlet ports and passages in proportion to the cross section of the cylinder. The least reduction in the density of the gas inside the combustion chamber will immediately create an inflow of combustible mixture, which will all but fill the vacuum. At high engine speeds, however, there is considerable resistance to the flow of the gaseous mixture through the carburetor, inlet pipe and valve ports, and the cylinder then receives considerably less than a full charge. By a full charge is meant a volume of charge at atmospheric density equal to the displacement of the piston.

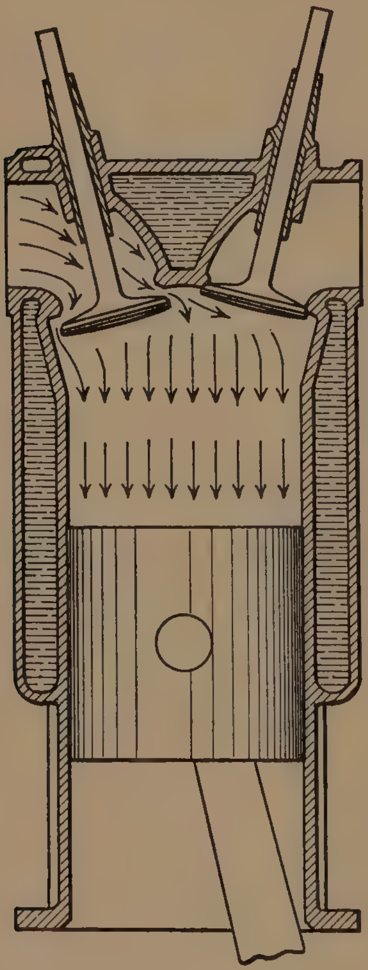


Fig. 2.—Inlet Stroke.

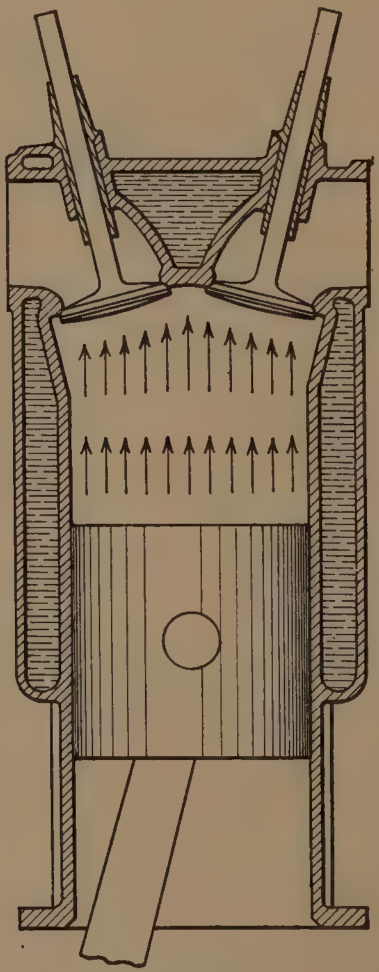


Fig. 3.—Compression Stroke.

Compression—Shortly after the completion of the inlet stroke, the inlet valve is closed, and during the rest of the return stroke of the piston the charge is compressed in the combustion chamber (Fig. 3). When the piston has reached the end of its return stroke—the compression stroke—the gas is compressed into from one-third to one-fifth or less of the volume it occupied at the beginning of the stroke. The space which it then occupies is known as the compression space. The objects of compressing the charge previous to ignition are as follows:

The average pressure in the cylinder during the power stroke increases with the pressure of the charge just prior to ignition, hence more power can be obtained from an engine using a comparatively high compression. It will be shown farther on in this chapter that the efficiency with which the heat of combustion is converted into mechanical work is increased by compression of the charge, and this gain in efficiency is due to the fact that with compression the charge can be carried through a wider temperature range in the cylinders during the power stroke.

Upon ignition the whole charge is immediately brought to a high temperature and a high pressure. By reason of its pressure the gas acts on the piston and thus converts heat energy into mechanical energy. However, owing to the contact of the highly heated gases with the relatively cool walls of the combustion chamber, heat is lost rapidly. This loss of heat, of course, represents a loss of energy, as the gases on cooling decrease in pressure and in their capacity for performing mechanical work. Other things being equal, this loss of heat is directly proportional to the area of the cooling surface with which a given quantity of charge is in contact. By compressing the charge before ignition the area of the surface with which it is in contact during ignition and expansion is lessened, and the heat losses are reduced. An engine in which the charge is compressed previous to ignition is, therefore, more efficient, transforming a greater percentage of the heat energy of the fuel into mechanical energy. In brief, compression increases the fuel efficiency and reduces the weight and bulk of the engine.

Ignition and Combustion—At or near the end of the compression stroke an electric spark is produced in the compression chamber which ignites the charge. If ignition of the whole charge took place instantaneously, the end of the stroke would be the proper time for the spark to occur. It has been shown, however, that it takes some time for the flame to be propagated through the charge, and for this reason, when the engine is running at relatively high speed the spark must occur slightly before the dead center for the best results.

Upon ignition the pressure suddenly rises to between four and five times what it was previously, and the piston is forced out under this pressure. The pressure multiplication factor varies with the composition of the charge, the form of the compression chamber and other variables. The actual pressure in a gasoline engine upon ignition is generally between 300 and 350 pounds per square inch, when the engine works at full power. The pressure drops, however, very

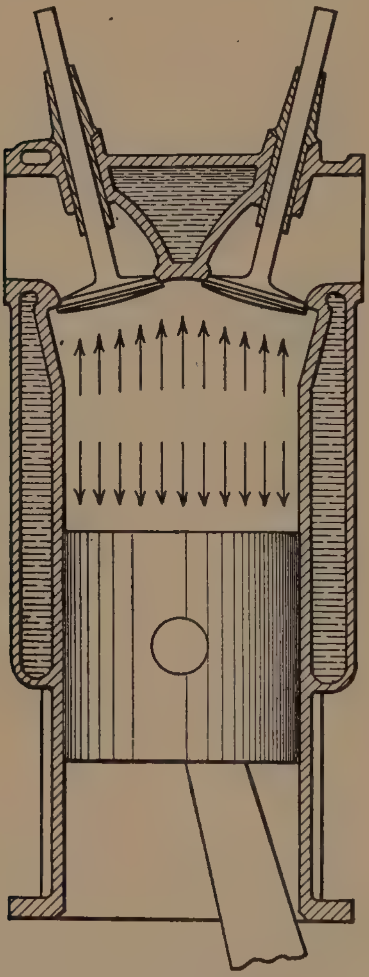


Fig. 4.—Power Stroke.

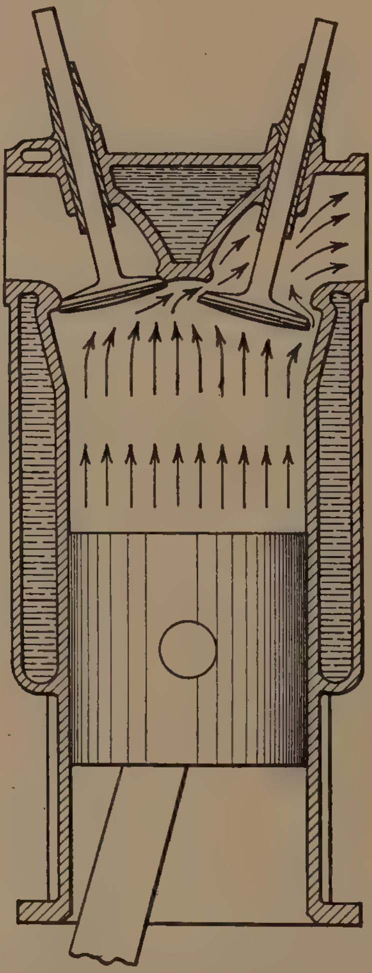


Fig. 5.—Exhaust Stroke.

rapidly, owing to the increase in the volume of the gases as the piston moves outward and to the abstraction of heat from the gases by the cylinder walls. When the power stroke (Fig. 4) is almost completed the exhaust valve begins to open and the spent gases, which are still under about 50 pounds pressure per square inch, escape rapidly.

Exhaust—The exhaustion of the burnt gases continues all through the following return stroke of the piston, the exhaust valve remaining open throughout this stroke (Fig. 5). During the first part of the exhaust period the gases are forced out by their proper pressure, which causes them to expand when no longer confined, but during the last portion of this stroke the piston practically sweeps the gases before it and pushes them out of the cylinder. The piston, however, does not sweep through the whole of the cylinder space and cannot clear the compression space of spent gases. This space remains always filled with the product of the previous combustion, and is for this reason also known as the dead space, or clearance space. At the end of the exhaust stroke, or slightly later, the exhaust valve closes, and at about the same time the inlet valve opens again and the cycle starts anew.

Diagram of Cycle—In the sketch, Fig. 6, is represented a cylinder of a gasoline engine with its piston in place. Into the piston head is secured a pipe leading to a device known as an indicator. This consists of a cylinder with a piston in it, with a coiled spring on top which tends to force the piston down to the bottom of the cylinder. The piston rod extends through the head of the cylinder and connects to a set of four links, joined so as to form a parallelogram. One member of this linkage is extended downwardly and is pivoted at its lower end to a fixed bracket. The top member is also extended and carries at its outer end a pencil which is lightly pressed against a paper card carried on a frame on top of the cylinder.

The indicator cylinder is always in direct communication with the engine cylinder, and the same gaseous pressure that acts on the engine piston acts on the indicator piston. The indicator piston will thereby be forced upward against the pressure of the spring. It is a law of coiled springs that when subjected to pressure they compress a distance which is directly proportional to the pressure. Hence, at any point in the stroke of the engine piston the indicator piston will be moved up in its cylinder a distance directly proportional to the pressure in the engine cylinder. The pencil will be moved a vertical distance directly proportional to the motion of the indicator piston, and hence the vertical height of the pencil is always a measure of the pressure acting in the engine cylinder. The pencil is also moved horizontally back and forth owing to the fact that it is carried by the engine piston, and its horizontal position, therefore, always corresponds to the position of the engine piston head; in other words, to the point in the stroke.

If the engine piston is moved back and forth in the cylinder while the valves are held open, the pressure in the cylinder will remain atmospheric, and the pencil will describe a straight horizontal line *A–A*, known as the atmospheric line. If, however, the engine performs its regular cycle and the inlet valve closes at the end of the inlet stroke, then during the following return stroke the gas will be compressed and the pencil will describe curve *A–B*, known as the compression curve. At the end of this stroke the charge is ignited and the pressure in the cylinder suddenly rises, in consequence of which the pencil describes a vertical or nearly vertical line *B–C*. Then, as the piston performs the next or power stroke, the pressure in the cylinder falls, first rapidly and then gradually more slowly, as indicated by the curved line *C–D*, which is known as the expansion curve. At the end of the power stroke, as the exhaust valve is opened, the pressure drops suddenly to almost atmospheric, as indicated by the short vertical line *D–A*.

These various straight and curved lines form together what is known as an engine diagram, which gives an accurate picture of what is going on inside the engine cylinder. Such a diagram cannot actually be taken from an automobile engine by the apparatus here shown, as the inertia of the moving parts would interfere with the results, but the sketch serves very well to explain what is meant by an indicator diagram.

Compression Calculation—At the present time it is customary to work at from 80 to 100 lbs. per sq. in. compression absolute. The higher the compression the greater the power obtained from an engine of given cylinder dimensions, and the higher the fuel ceonomy, but when the compression exceeds a certain limit there is danger of trouble from overheating of the cylinder, self-ignition (ill-timed) due to the heat of the compression, and knocking or detonation. With very small cylinders it is possible to use a somewhat higher compression than with larger ones, owing to the greater ratio of cooling area to cylinder volume in the former.

The compression of the charge approximates an adiabatic compression, the expression for which, as we have seen, is

$$PV^\gamma = \textit{Constant.}$$

If P_1 be the pressure in the cylinder when the compression stroke begins; V_1, the volume occupied by the gases at the beginning of the compression stroke (volume swept through by piston plus compression space volume); P_2, the pressure at the completion of the compression stroke and V_2 the volume occupied by the gases at the completion of that stroke (compression space volume), then, evidently,

$$P_1V_1^\gamma = P_2V_2^\gamma = \textit{Constant.}$$

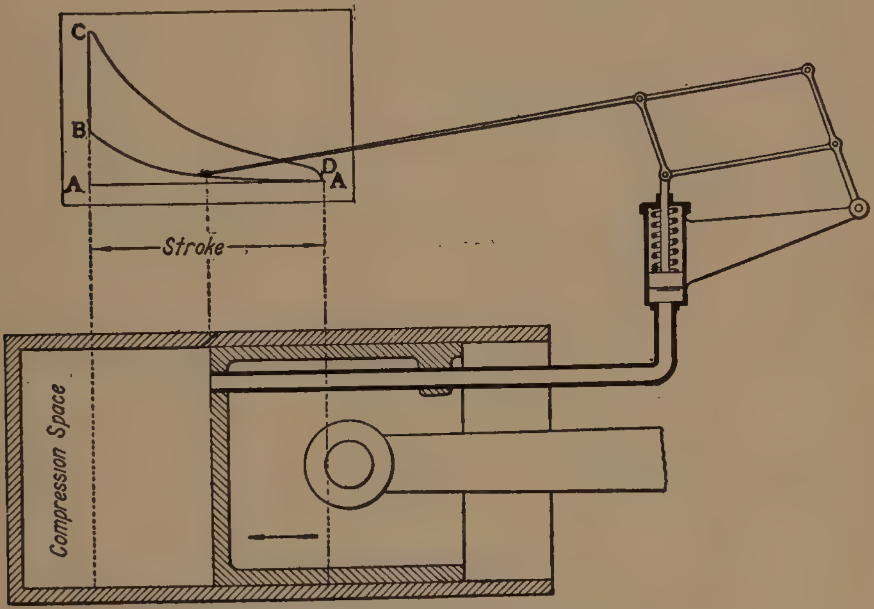


FIG. 6.—RECORDING CYLINDER PRESSURE CHANGES.

From which it follows that the compression pressure

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^\gamma.$$

The initial pressure P_1 varies with the piston speed, the relative size and form of the passages, and with other factors. At low engine speeds it is generally very close to atmospheric pressure or 14.7 pounds per square inch, while at normal engine speeds this pressure is not more than from 12 to 13 pounds per square inch. The ratio V_1/V_2 is known as the compression ratio.

For adiabatic compression the exponent γ is equal to 1.405, but when the charge is compressed in an engine cylinder some heat is lost to the cylinder walls and compression then is not adiabatic but polytropic. Measurements on actual indicator diagrams and an analysis of the results has shown that γ has an average value of 1.3. Assuming, for example, an initial pressure of 13 lbs. per. sq. in. and a compression ratio of 4 to 1, we find the compression pressure to be

$$13 \times 4^{1.3} = 78.8 \text{ lbs. per sq. in., abs.}$$

Chart I herewith instantly gives the compression pressure for any initial pressure from 12 lbs. per sq. in., and any compression ratio from 3 to 5.

It is obvious that the compression pressure of an engine of given construction is not fixed, but varies with the speed, since the speed affects the degree of filling, and, hence, the initial pressure. In giving the specifications of an engine it is better, therefore, to give the compression ratio—the ratio of the volume of piston sweep plus the compression volume, to the compression volume—as this is fixed.

Work of Compression—The compression of the charge in the compression chamber requires the expenditure of a certain amount of work. Let

b = Cylinder bore in inches;
l = Length of stroke;
r = Ratio of compression.

Then, considering the compression space in the form of an extension of the cylinder, its height will be

$$\frac{l}{r-1}.$$

The length from the cylinder head to the piston head when the latter is at the beginning of the compression stroke is

$$l + \frac{l}{r-1} = \frac{rl}{r-1}.$$

Suppose that at the beginning of the compression stroke the pressure in the cylinder is P_i. The total pressure against the piston from the inside of the cylinder then is $\pi b^2 P_1/4$.

Now consider the piston to be at a distance x from the cylinder head (Fig. 7). The internal pressure then is

$$P_i\left(\frac{\frac{rl}{r-1}}{x}\right)^{1.3} = P_i\left(\frac{rl}{r-1}\right)^{1.3} x^{-1.3}.$$

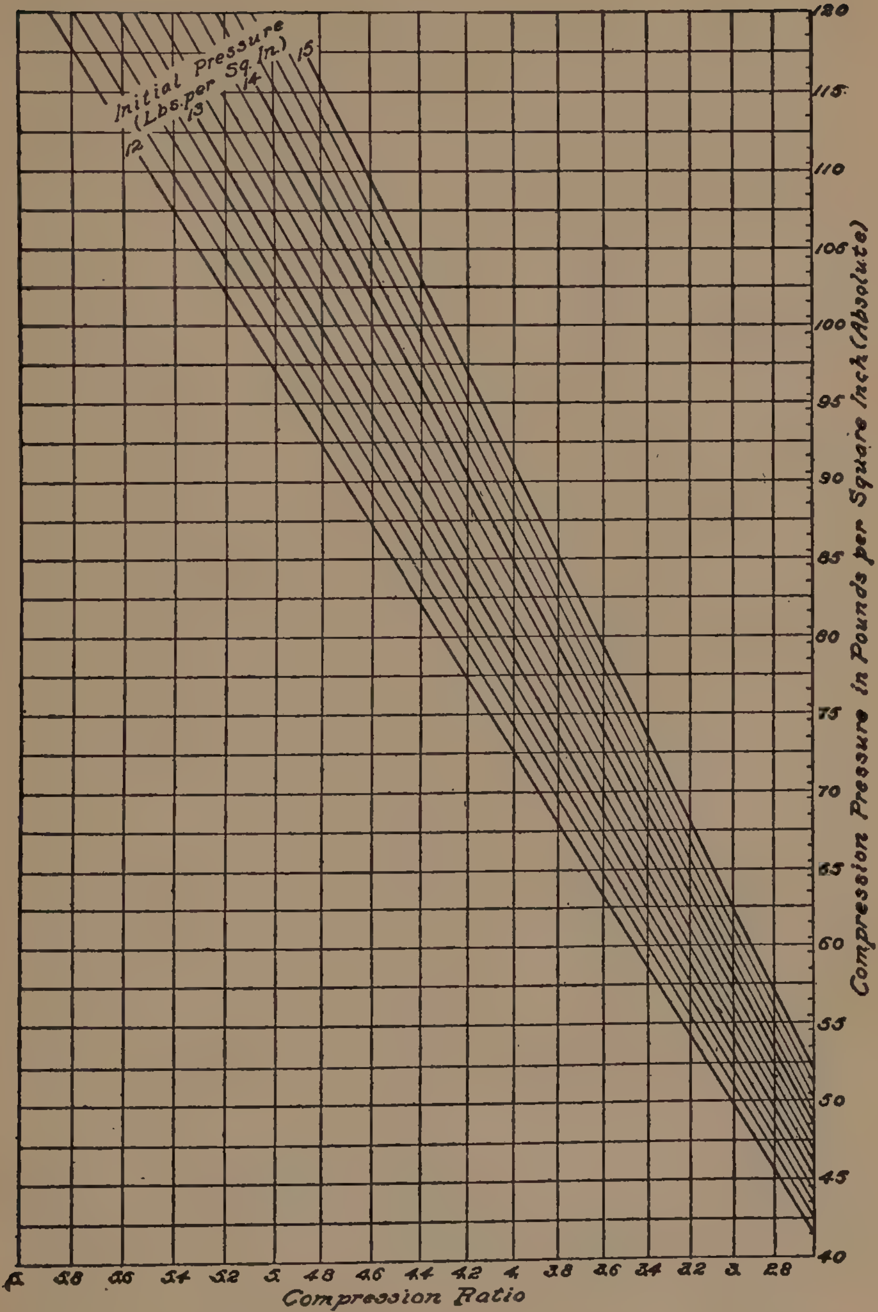


Chart I.—Compression Pressures for Various Initial Pressures and Compression Ratios.

If the piston is now moved an infinitesimal distance $-\,dx$ to the left against this pressure, the work done will be

$$dW_c = -\frac{\pi b^2}{4} \times P_t\left(\frac{rl}{r-1}\right)^{1.3} x^{-1.3}dx.$$

This is a differential equation of the form

$$dy = ax^m dx,$$

the integral of which is

$$y = a\frac{x^{m+1}}{m+1} + Constant.$$

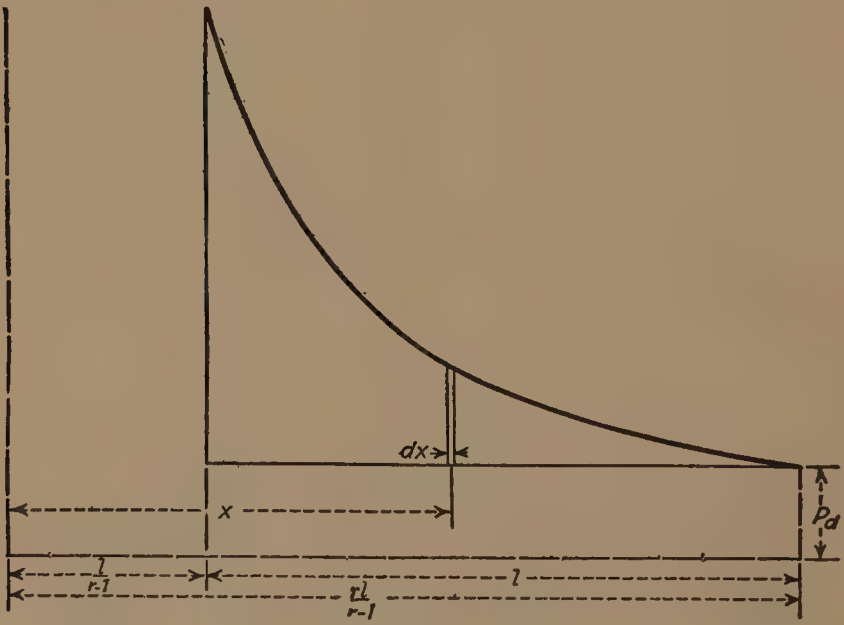


FIG. 7.

However, since we integrate in this case between limits, the constant vanishes. The limiting values of x are

$$\frac{l}{r-1} \quad \text{and} \quad \frac{rl}{r-1}.$$

Hence

$$\int_{rl/(r-1)}^{l/(r-1)} -\frac{\pi b^2}{4}P_t\left(\frac{rl}{r-1}\right)^{1.3} x^{-1.3}dx$$

$$= \frac{\pi b^2}{4} P_i \left(\frac{rl}{r-1}\right)^{1.3} \frac{1}{0.3}\left[\left(\frac{l}{r-1}\right)^{-0.3} - \left(\frac{rl}{r-1}\right)^{-0.3}\right]. \quad . \quad (8)$$

Since

$$\left(\frac{rl}{r-1}\right)^{1.3}\left(\frac{rl}{r-1}\right)^{-0.3} = \frac{rl}{r-1},$$

multiplying the part outside the square brackets by

$$\left(\frac{rl}{r-1}\right)^{-0.3},$$

and dividing the part inside the brackets by the same expression, we have

$$W_c = \frac{\pi b^2}{4} \frac{P_i}{0.3}\left(\frac{rl}{r-1}\right)\left[\left(\frac{1}{r}\right)^{-0.3} - 1\right],$$

which may be simplified to

$$W_c = \frac{\pi b^2 P_i}{1.2}\left(\frac{rl}{r-1}\right)(r^{0.3} - 1).$$

This equation gives the result in inch-pounds, if the pressure P_i is given in pounds per square inch and b and l are given in inches. In order to get it in the more usual foot-pounds, we divide by 12, which gives

$$W_c = \frac{\pi b^2 P_i}{14.4}\left(\frac{rl}{r-1}\right)(r^{0.3} - 1) \text{ ft.-lbs.} \quad . \quad . \quad . \quad (10)$$

This, of course, does not take account of the fact that the atmosphere presses with a pressure of 14.7 pounds per square inch against the outside of the piston, and thus helps to compress the gas. The amount of work given by the equation would be required to compress the charge if the cylinder were placed in a vacuum. It is permissible to neglect the effect of the atmospheric pressure, because the energy thus derived during the compression stroke must be expended again during the following expansion stroke. However, if it is desired to determine the net work of compression, this can easily be done by subtracting from the result of equation (10) the energy derived from the atmosphere during the compression stroke, which is

$$\frac{\pi b^2 l 14.7}{4 \times 12} \text{ ft.-lbs.}$$

Work of Expansion—If after the completion of the compression stroke the charge is allowed to expand again, the work done by the expansion will be the same as that expended in compressing the charge. Equation (10) would therefore apply to the expansion as well as to the compression if the gas were expanded in the same state in which it was compressed. However, at the end of the compression stroke the charge is burned, whereby its pressure is multiplied several times. Calling this pressure ratio a, we may write for the work of expansion:

$$W_e = \frac{\pi b^2 a P_t}{14.4}\left(\frac{rl}{r-1}\right)(r^{0.3} - 1) \text{ ft.-lbs.}$$

The difference of the work of expansion and the work of compression is evidently the useful work of one cycle (neglecting the slight loss due to suction and exhaust back pressure). The useful work of one cycle may therefore be represented by the following equation:

$$W_U = \frac{\pi b^2 l P_t}{14.4}(a-1)\left(\frac{r}{r-1}\right)(r^{0.3} - 1) \text{ ft.-lbs.}$$

$$= \frac{\pi b^2 l P_t}{14.4}(a-1)\left(\frac{r^{1.3} - r}{r-1}\right) \text{ ft.-lbs.}$$

The displacement of the piston during one stroke is

$$D = \frac{\pi b^2 l}{4} \text{ cu. ins.}$$

and substituting this factor in the above equation we get

$$W_U = \frac{DP_t}{3.6}(a-1)\left(\frac{r^{1.3} - r}{r-1}\right) \text{ ft.-lbs.} \quad . \ . \ . \ . \quad (11)$$

The area of the indicator diagram represents useful work done by the gases on the piston during the power stroke. Owing to the fact that in an actual diagram all of the corners are cut off the ideal diagram as represented by Fig. 8, the actual diagram does not represent as much work as the ideal diagram, and it is therefore necessary to introduce a correction factor, known as the diagram factor, in equation (11) for the useful work of one cycle. This diagram factor, which is usually assumed to be equal to 0.9 or thereabouts, takes account of the losses occasioned by the non-instantaneous supply of heat to the charge, of the early opening of the exhaust valve and of the fluid resistance encountered during the inlet and exhaust strokes. This latter loss is represented by the area enclosed between the inlet and exhaust lines of the diagram.

Negative work is being done whenever the piston moves in opposition to the direction of gas pressure upon it.

Introducing the diagram factor f in equation (11) we get

$$W_U = \frac{DP_i f}{3.6}(a-1)\left(\frac{r^{1.3}-r}{r-1}\right). \quad . \quad . \quad . \quad . \quad (12)$$

The work per cubic inch of displacement is therefore

$$w_U = \frac{P_i f}{3.6}(a-1)\left(\frac{r^{1.3}-r}{r-1}\right) \text{ ft.-lbs.} \quad . \quad . \quad (13)$$

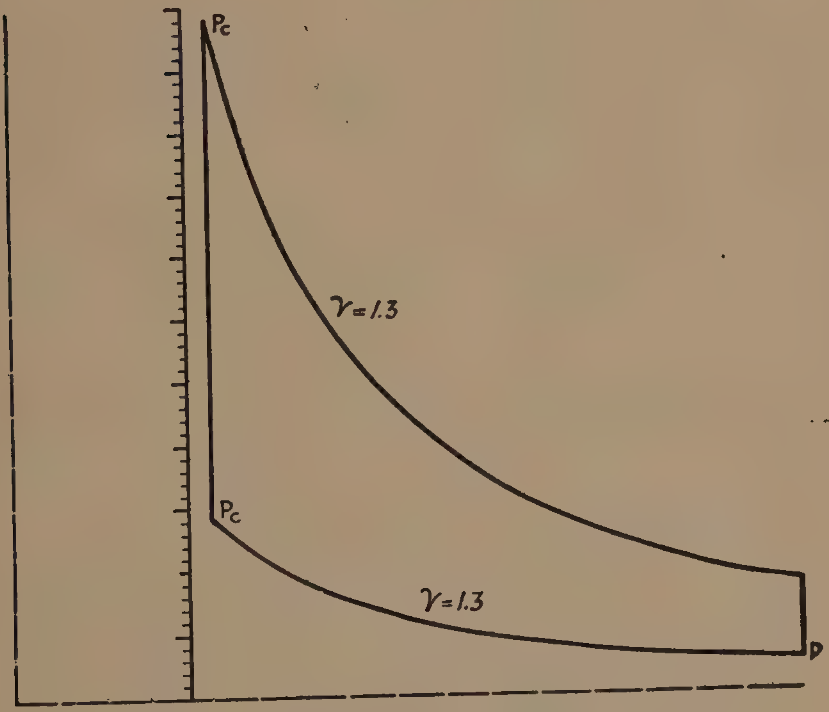


FIG. 8.—IDEAL DIGRAM.

To get the mean effective pressure in the cylinder during the power stroke we merely multiply this value by 12 to convert it to inch-pounds, for it is obvious that with a mean effective pressure p on a piston head area of one square inch, and a piston motion of one inch (one cubic inch displacement) we get p inch-pounds of work. Hence,

$$\text{M.E.P.} = \frac{12P_i f}{3.6}(a-1)\left(\frac{r^{1.3}-r}{r-1}\right) \text{ lbs. per sq. in.} \quad . \quad . \quad (14)$$

$$= 3.33P_i f(a-1)\left(\frac{r^{1.3}-r}{r-1}\right) \text{ lbs. per sq. in.}$$

To determine the torque (turning moment or crank moment) of the engine we multiply the indicated work W by the mechanical efficiency η. The energy of one power stroke (in each cylinder in the case of a multi-cylinder engine) furnishes torque for two complete revolutions of the crankshaft. With a torque T on the crankshaft the work done during two revolutions is $4\pi T$, and this must be equal to the product of the useful indicated work W by the mechanical efficiency η—

$$4\pi T = \frac{D\eta P_i f}{3.6}(a - 1)\left(\frac{r^{1.3} - r}{r - 1}\right) \text{ ft.-lbs.}$$

Hence

$$T = \frac{D\eta P_i f}{45.2}(a - 1)\left(\frac{r^{1.3} - r}{r - 1}\right) \text{ lb.-ft.,} \quad . \quad . \quad (15)$$

and the torque per cubic inch displacement

$$t = \frac{\eta P_i f}{45.2}(a - 1)\left(\frac{r^{1.3} - r}{r - 1}\right) \text{ lb.-ft.} \quad . \quad . \quad . \quad (16)$$

The term involving the compression ratio r has the following values for different values of r:

$r =$	3.00	3.25	3.50	3.75	4.00	4.25
	0.585	0.612	0.638	0.664	9.688	0.711
$r =$	4.50	4.75	5.00	5.25	5.50	5.75
	0.733	0.754	0.775	0.796	0.816	0.835

Remembering that the useful work per cycle increases directly as the value of this term, it becomes obvious that the compression ratio has a strong effect on the power output of an engine.

A factor that is much used in the discussion of engine characteristics is the brake mean effective pressure. This is equal to the indicated mean effective pressure multiplied by the mechanical efficiency η and is convenient because it can be readily calculated from the result of torque or horse power measurements made on the engine. The brake mean effective pressure is evidently

$$\text{B.M.E.P.} = 3.33 P_i f \eta (a - 1)\left(\frac{r^{1.3} - r}{r - 1}\right) \text{ lbs. per sq. in.} \quad (17)$$

It will be seen that this last expression bears a great deal of resemblance to that for the torque per cubic inch displacement; in fact, the two factors are proportional, the brake mean effective pressure being equal to 150.8 times the torque per cubic inch displacement. There is a similar relation between the brake mean effective pressure and the torque of the whole engine—

$$\text{B.M.E.P.} = 150.8\frac{T}{D}. \quad \ldots\ldots \quad (18)$$

Exhaust and Suction Curves—Fig. 8 represents ideal compression and expansion curves. The actual curves differ slightly from these, for several reasons. In the first place, not all of the gas in the combustion chamber is ignited instantly, and, therefore, we do not have in an actual diagram a straight vertical line ${}_eP_cP$ making very distinct junctions with the compression and expansion curves. If the ignition spark is produced while the piston is at the end of the stroke the line ${}_eP_cP$ will be slightly inclined to the right, but in order to convert the maximum percentage of the heat energy of the charge into useful work the spark must occur a moment before the piston reaches the end of the stroke. The result of this is that toward the end of the compression stroke the compression curve assumes a rising inflection, as indicated in Fig. 9, and instead of joining the vertical ignition line sharply, it does so with a curve of considerable radius. The junction between the explosion and expansion lines will be rounded off, instead of being pointed, and the maximum pressure attained will be relatively lower than indicated in Fig. 8.

Another thing which modifies the form of the diagram is the fact that the expansion cannot be continued in the cylinder till the end of the expansion stroke and the exhaust valve remain closed. It must be remembered that the exhaust valve does not open instantly, but gradually, and if it began to open only at the end of the stroke, the piston upon the beginning of the exhaust stroke would encounter a back pressure practically equal to the pressure which was acting upon it just before the completion of the power stroke. There would be a very considerable back pressure during a large portion of the return stroke, and this would not only cause a very considerable loss in power, but, owing to the retention of the hot gases, would tend to overheat the cylinder. For this reason the exhaust valve begins to open when about seven-eighths of the power stroke is completed, and the pressure in the cylinder then drops more rapidly than it otherwise would. This takes the upper right-hand corner off the diagram, as shown by line *D E*, Fig. 9.

So far we have considered only the curves traced by the pencil of the indicator during the compression and expansion strokes. When the piston begins the exhaust stroke, and, indeed, throughout that stroke, the pressure in the cylinder is somewhat above atmospheric and the indicator pencil will trace a line, slightly curved, a little above the atmospheric line. When the engine runs at relatively low speed the exhaust line will join the atmospheric line at the end of the stroke, but at high speed it will then still be above the atmospheric line, and for this reason the exhaust valves in high-speed engines are held open a little longer than the end of the exhaust stroke. During the first part of the following inlet stroke the pressure in the cylinder drops fairly rapidly, partly because some more of the burned gases escape and partly because the space occupied by the gases rapidly increases. A short distance from the beginning of the stroke the pressure drops below atmospheric, and from that point on to the end of the stroke there is a suction or a partial vacuum in the cylinder, the suction line lying below the atmospheric line and being generally slightly concaved toward the atmospheric line. At high speed the pressure is still below atmospheric at the end of the suction stroke, and the inlet valve of a high-speed engine remains open for a short time after the piston begins the compression stroke, because the remaining difference in pressure inside and outside the cylinder insures a continued inflow. From the beginning

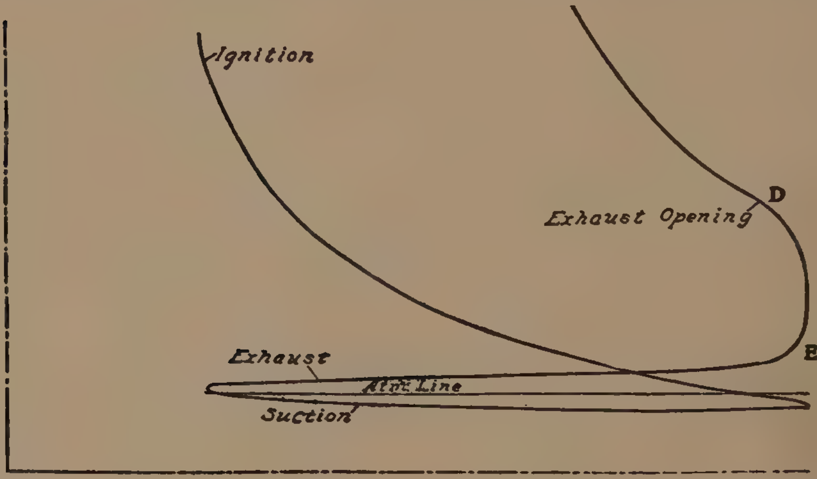


FIG. 9.—LOW PRESSURE PORTION OF DIAGRAM ON ENLARGED SCALE.

of the compression stroke the pressure in the cylinder rises. The pressure line soon cuts the atmospheric line, and then, as the compression proceeds, rises at a constantly increasing rate, as already explained.

When the engine runs at relatively low speed the pressure in the cylinder is atmospheric at the beginning and end of the suction stroke. The pressures during the exhaust and suction stroke are so small, as compared with the pressure of explosion, that it is necessary to draw this part of the diagram to a different scale from the compression and expansion curves, in order to show it plainly.

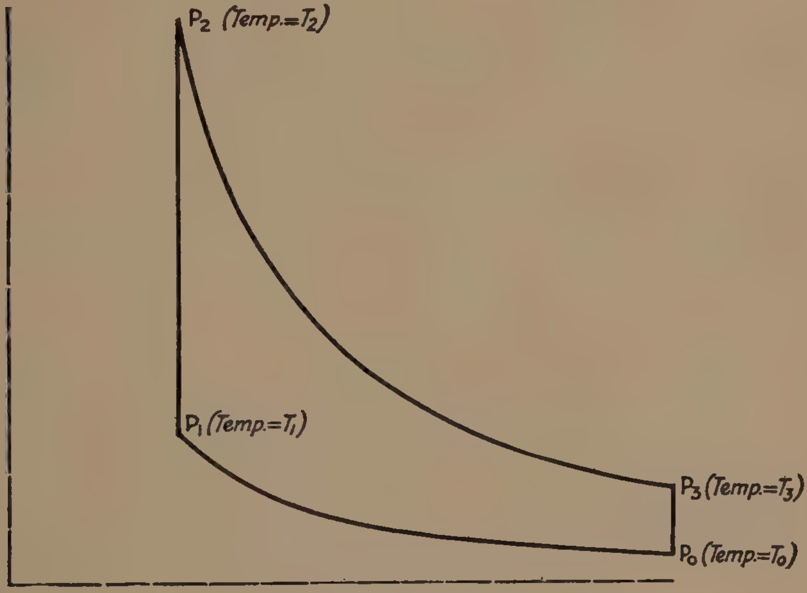


FIG. 10.

Thermal Efficiency—The thermal efficiency is the ratio of the thermal energy actually utilized in the engine to the total amount of heat energy supplied to the gases. In the form of an equation this may be written:

$$\eta = \frac{\textit{Energy supplied} - \textit{energy discharged}}{\textit{energy supplied}}$$

Referring to Fig. 10 let

T_1 = Absolute temperature of the gas just previous to ignition;
T_2 = Absolute temperature of the gas after ignition;

T_3 = Absolute temperature at the completion of the expansion stroke;

T_0 = Absolute temperature of the gas as it is drawn into the cylinder;

The heat energy being supplied to the gas at constant volume, its value—considering a unit quantity of gas—is evidently $C_v(T_2 - T_1)$. The energy is also discharged at constant volume at the end of the expansion stroke, the amount so discharged being $C_v(T_3 - T_0)$. We may write, therefore,

$$\eta = \frac{(T_2 - T_1)C_v - (T_3 - T_0)C_v}{(T_2 - T_1)C_v} = 1 - \frac{T_3 - T_0}{T_2 - T_1}. \quad (19)$$

But since both the compression and expansion are adiabatic, for which form of compression or expansion

$$TV^{\gamma-1} = \textit{Constant},$$

we may write

$$\frac{T_1}{T_0} = \frac{T_2}{T_3} = \left(\frac{V_0}{V_1}\right)^{\gamma-1};$$

hence

$$T_3 = \frac{T_2}{T_1}T_0,$$

which substituted in (19), gives

$$\eta = 1 - \frac{\frac{T_2}{T_1}T_0 - T_0}{T_2 - T_1}.$$

Dividing both the numerator and denominator of the fraction by

$$\frac{T_2}{T_1} - 1:$$

$$\eta = 1 - \frac{T_0}{T_1}$$

$$= 1 - \left(\frac{V_1}{V_0}\right)^{\gamma-1}$$

$$= 1 - \left(\frac{1}{r}\right)^{\gamma-1}$$

By assigning to γ its value of 1.405 we obtain what is known as the air-cycle efficiency, which is often referred to in theoretical discussions. It has been pointed out already that in an actual engine the value of γ is lower owing to losses of heat to the cylinder wall.

Since in gasoline engine work γ is always in the neighborhood of 1.3, we may write

$$\eta = 1 - \left(\frac{1}{r}\right)^{0.3}, \quad \ldots \ldots \quad (20)$$

which simple equation gives the theoretical thermal efficiency of the Otto cycle gas engine. This equation is of interest in that it strikingly shows the influence of the compression ratio on the thermal efficiency. Thus, for instance, with a compression ratio of 3 the theoretical efficiency is

more power

$$1 - (\tfrac{1}{3})^{0.3} = 0.281,$$

and with a compression ratio of 4

$$1 - (\tfrac{1}{4})^{0.3} = 0.341$$

6 to 7 21.4%

4.11% 12.3

and with a compression ratio of 5

$$1 - (\tfrac{1}{5})^{0.3} = 0.383.$$

Fuel Consumption and Thermal Efficiency.—It has been pointed out that one brake horse power is equal to 33,000 foot-pounds per minute, and it is therefore equal to

$$60 \times 33{,}000 = 1{,}980{,}000 \text{ ft.-lbs. per hr.}$$

Hence,

$$1 \text{ hp.-hr.} = 1{,}980{,}000 \text{ ft.-lbs.}$$

Now let a be the fuel consumption in pounds per horse power-hour and H the heat value of this fuel in B.T.U. per pound. Then, since one B.T.U. is equal to 778-ft.-lbs., the mechanical equivalent of this heat value is

$$a \times H \times 778 \text{ ft.-lbs.},$$

and the thermal efficiency, which is equal to the ratio of the mechanical work done to the heat energy of the fuel consumed, is

$$\eta = \frac{1{,}980{,}000 \times 100}{a \times H \times 778} = \frac{254{,}500}{a \times H} \text{ per cent.}$$

For an ordinary gasoline with a heat value of 19,000 B.T.U. per pound this reduces to

$$\eta = \frac{13.4}{a} \text{ per cent.}$$

Application of Formulae—We will now work out a practical example with the aid of the formulae developed in the foregoing. We will assume a cylinder with a bore $b = 3$ in. and a stroke $l = 4\tfrac{1}{2}$ in. Let the compression ratio r be 4.5

[illegible] avg per .54/HP hr.

The normal speed of passenger car engines with cylinders of these dimensions is usually about 3000 revolutions per minute. It has been found that at such a speed an engine with large size valves properly timed draws in about 65 per cent of its displacement volume of charge per cycle. That no full charge is drawn in is due to the fact that, owing to "wire drawing" in the valve passages, the pressure in the cylinder will not be reduced to atmospheric at the end of the exhaust stroke and will not be up to atmospheric at the end of the inlet stroke while, in addition, the charge will be expanded by the heat supplied to it in the inlet manifold. We may assume that the pressure at the beginning of the compression stroke is 12 lbs. per sq. in. The compression pressure than will be

$$12 \times 4.5^{1.3} = 85 \text{ lbs. per sq. in.}$$

absolute, or 70 lbs. per sq. in. gauge.

The mechanical efficiency of the engine may be taken at 80 per cent. $= 0.80$), and the diagram factor f at 0.9, while the ratio of pressure multiplication on explosion will be about 4.5, absolute pressures being figured with. The value of the term $(r^{1.3} - r)/(r - 1)$, which occurs in all of the formulae, is 0.733 for $r = 4.5$. The piston displacement of a 3 by 4.5 in. cylinder is 31.8 cu. in. The indicated work per cycle (equation 12) is

$$W_u = \frac{31.8 \times 12 \times 0.9}{3.6} \times 3.5 \times 0.733 = 244.7 \text{ ft.-lbs.,}$$

and the indicated work per cycle per cubic inch displacement,

$$w_u = \frac{244.7}{31.8} = 7.7 \text{ ft.-lbs.}$$

Since at 3000 r.p.m. there are 1500 explosions in the cylinder per minute, the work done per minute is

$$1500 \times 244.7 = 367{,}050 \text{ ft.-lbs.}$$

Now, one mechanical horse power is equal to 33,000 ft.-lbs. per minute, so this 3 by 4.5 in. cylinder develops

$$\frac{367{,}050}{33{,}000} = 11.12 \text{ indicated horse power.}$$

Since we have assumed the mechanical efficiency to be 80 per cent, the brake horse power is

$$0.80 \times 11.12 = 8.9$$

A four cylinder engine of this size would therefore develop 35.6 hp. and a six cylinder, 53.4 hp.

The mean effective pressure (equation 14) is

$$3.33 \times 12 \times 0.9 \times 3.5 \times 0.733 = 91.6 \text{ lbs. per sq. in.}$$

and the brake mean effective pressure,

$$0.80 \times 91.6 = 73.3 \text{ lbs. per sq. in.}$$

The torque for a single cylinder would be (equation 15),

$$T = \frac{31.8 \times 12 \times 0.9 \times 0.80}{45.2} \times 3.5 \times 0.733 = 15.6 \text{ lb.-ft.},$$

and the torque per cubic inch displacement,

$$t = \frac{15.6}{31.8} = 0.49 \text{ lb.-ft.}$$

The volumetric efficiency has been set at 65 per cent, hence, the amount of air drawn in per hour by a six cylinder engine of this size is

$$\frac{3000 \times 3 \times 0.65 \times 31.8 \times 60}{1728} = 6456 \text{ cu. ft.}$$

and since one cubic foot of air at normal atmospheric pressure and temperature weighs 0.076 lb., this air weighs,

$$6456 \times 0.076 = 490 \text{ lbs.}$$

Assuming the mixture ratio to be 15 : 1, this air will be mixed with

$$490/15 = 32.7 \text{ lbs. of gasoline.}$$

Since the six cylinder engine develops 53.4 brake horse power, the fuel consumption is at the rate of

$$32.7/53.6 = 0.61 \text{ lb. per hp.-hr.}$$

The values above calculated do not represent the best, but fair average performance of modern engines.

CHAPTER IV

Conversion of Reciprocating into Rotary Motion

In power plant engineering, if simplicity and low cost of manufacture are the most essential qualities, it is usually best to use the single cylinder form of construction. All of the early stationary gas engines were of that type, and farm engines and other small power units operating on the internal combustion principle even today have only a single cylinder. In automotive work, however, other considerations predominate, and for automobiles, engines with as many as twelve cylinders have been used, while in aircraft work the number of cylinders has been carried up to eighteen. In the United States, four is the minimum number of cylinders used on motor vehicles other than motorcycles. The reasons which induced automotive designers to accept the complication of multiple cylinder construction may be explained as follows:

It has already been pointed out that in a four-stroke engine only one stroke in every four is a power stroke; between succeeding power strokes there are three so-called idle strokes, and during one of these, the compression stroke, power is even consumed by the engine. To keep the engine running at a fairly uniform speed against a constant or nearly constant resistance, it is necessary to employ a heavy flywheel in which some of the energy liberated is stored during the expansion stroke, to be given out again during the three following idle strokes. But a heavy flywheel is not desirable on a car, because of the additional weight, which consumes fuel, lessens the speed and hastens the wear of the tires.

Another objectionable feature of the single cylinder engine is its lack of balance. Vibrations in an engine are due to the reactions of the explosion and of the inertia forces. Now, if we consider an engine of a given output, the force of individual explosions is necessarily much greater in a single-cylinder than in a multicylinder design. Moreover, in a single-cylinder engine the entire reciprocating mass is in a single unit, and the reactions of its inertia forces produce a strong vibrating effect, while in a multicylinder engine the reciprocating mass can be divided into several units so arranged as to move in opposite directions at any given moment, thereby neutralizing the inertia effects.

The reciprocating motion of the piston is converted into rotary motion of the crankshaft by means of an intermediate connecting rod. The connecting rod is hinged to both the piston and the crankshaft. Its upper end reciprocates in harmony with the piston, while its lower end rotates with the crank pin. Before proceeding with a discussion of crank and piston motion it may be well to state a few fundamental principles which apply to these problems.

Fundamental Relations—Speed is the quotient of a distance traveled by the time occupied. If the speed is constantly varying, its momentary value can be found by dividing the distance dx traveled in an infinitesimal time dt by dt. That is, the speed or velocity

$$v = \frac{dx}{dt}.$$

Acceleration is the quotient of a velocity by the time in which it was acquired. If the acceleration is not constant it is necessary in determining its value to take an infinitesimal increase in velocity dv and divide it by the time dt in which this increase in velocity accrued. Hence acceleration

$$a = \frac{dv}{dt} = \frac{d^2x}{dt^2}.$$

The acceleration of a body requires the expenditure of energy. A body in motion has stored up in it a certain amount of energy, called its kinetic energy, or *vis viva,* and this energy is, of course, imparted to it when it is set in motion. Gravity, which acts on a body with a force equal to its weight, imparts a velocity of 32.16 feet per second to it in one second. The unit of velocity is one foot per second, though a unit of one foot per minute is also used. The unit of acceleration is one foot per second per second. The unit of force generally employed in mechanical work in this country is the pound. Then, since a unit of force acting on a unit mass (one pound) accelerates it 32.16 feet per second per second, and since the acceleration is directly proportional to the acting force and inversely proportional to the mass acted upon, we may write

$$a = \frac{32.16F}{W}, \qquad \text{or} \qquad F = \frac{Wa}{32.16}.$$

The factor 32.16 is known as the acceleration of gravity, and is usually denoted by g.

Piston Motion—In studying the motions of the different parts of an engine, it is generally assumed that the crank pin rotates at a uniform speed. This supposition is nearly correct. A typical arrangement of the piston, connecting rod and crankshaft is shown in Fig. 11, and in Fig. 12 these moving parts are represented diagrammatically. The line $l/2$ represents the crank-arm and the line nl the connecting rod. The lengths of these lines represent the "center to center" distances; that is, in the case of the crank-arm the distance between the center of the crankshaft journal and the center of the crank-pin, and in the case of the connecting rod the distance between the center of the crank-pin bearing and the center of the piston-pin bearing. It will be seen that in Fig. 12 the crank-arm has moved from the upper dead center position through an angle θ, and the piston has in consequence moved down a distance x from its topmost position.

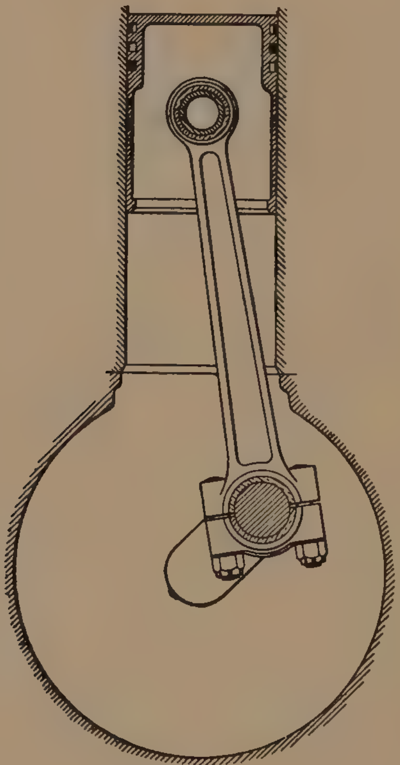


FIG. 11.

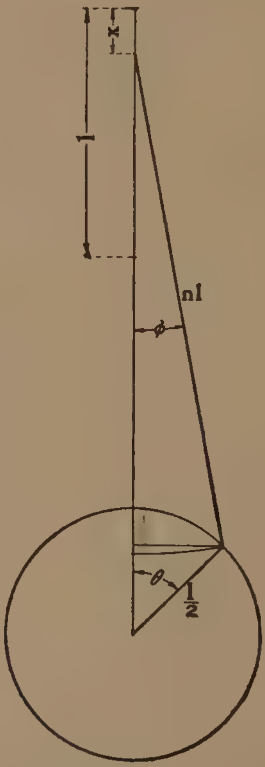


FIG. 12.

[illegible]

It is now desired to find the relation between θ and x, so that we may calculate the position of the piston corresponding to any position of the crank, as well as the speed and acceleration of the piston corresponding to any crank position and a given speed of rotation of the crankshaft.

Referring now to Fig. 12, it is obvious that

$$x = \frac{l}{2}(1 - \cos\theta) + nl(1 - \cos\phi)^{*}. \quad . \quad . \qquad (21)$$

But

$$\cos\phi = \sqrt{1 - \sin^2\phi}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (22)$$

and

$$nl\sin\phi = \frac{l}{2}\sin\theta, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (23)$$

so that

$$\sin\phi = \frac{\sin\theta}{2n},$$

which substituted in (22) gives

$$\cos\phi = \sqrt{1 - \frac{\sin^2\theta}{4n^2}}.$$

Inserting this value of $\cos\phi$ in equation (21) we get

$$x = \frac{l}{2}(1 - \cos\theta) + nl\left(1 - \sqrt{1 - \frac{\sin^2\theta}{4n^2}}\right). \quad . \qquad (24)$$

This equation enables us to determine the portion of the stroke completed while the crank is in any position. Now assume that the ratio n of the connecting rod length to length of stroke is 2, and let the length of stroke be unity. Then, after a quarter turn, when $\theta = 90°$, the fraction of the stroke completed will be

$$\tfrac{1}{2} + 2(1 - \sqrt{1 - \tfrac{1}{16}} = 0.564.$$

---

* For the third and fourth quadrants, i. e., the return stroke, if angles ϕ to the left of the axis are considered negative, the second half of this expression should be preceded by a $-$ instead of a $+$ sign. As here given the equation applies to the down stroke only, but the return stroke is a reversed duplicate of this.

It is thus seen that when the crank has turned through one-quarter of a revolution, more than half of the stroke is completed. The piston, therefore, travels faster during the first half of its down stroke than during the last half; similarly it may be shown that it travels slower during the first half of the return stroke than during the last half. In Fig. 13 the piston is shown in the maximum speed position, with the connecting rod and crank arm in the corresponding positions for both the down stroke and the return stroke.

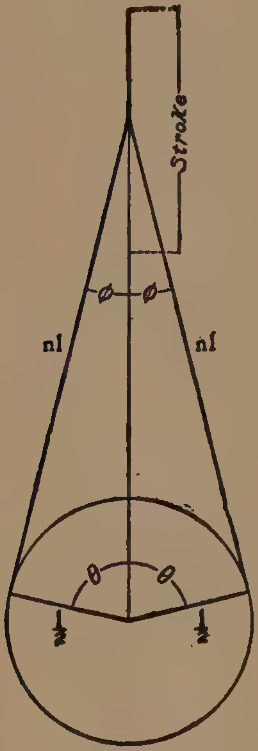


FIG. 13.

Piston Speed.—The differentiation of equation (24) with the object of deriving an expression for the instantaneous piston speed involves some difficulties. In fact, the right-hand member of the equation can be differentiated only after being developed into a series —a rather complicated proceeding. But the difficulty can be circumvented without serious error by adding to the expression under the radical the term

$$\frac{\sin^4 \theta}{64n^4}.$$

This makes that expression a perfect square, and, taking the root, we have

$$\sqrt{1 + \frac{\sin^4 \theta}{64n^4} - \frac{\sin^2 \theta}{4n^2}} = 1 - \frac{\sin^2 \theta}{8n^2}.$$

Substituting this in equation (24), we get

$$x = \frac{l}{2}(1 - \cos \theta) + nl\left(\frac{\sin^2 \theta}{8n^2}\right). \quad . \quad . \quad . \quad . \quad (25)$$

Differentiating, we find for the momentary piston speed

$$v = \frac{dx}{dt} = \frac{l}{2} \sin \theta \frac{d\theta}{dt} + \frac{l}{4n} \sin \theta \cos \theta \frac{d\theta}{dt}.$$

But since the angular speed of the crank is constant

$$\frac{d\theta}{dt} = \frac{\theta}{t} = \frac{2\pi N}{60},$$

radians per second, N being the number of revolutions per minute of the crank. Substituting this value and dividing by 12 to get the result in feet, we have

stroke in in

$$v = \frac{\pi l N}{720}\left(\sin\theta + \frac{1}{2n}\sin\theta\cos\theta\right) \text{ ft. per sec.} \quad . \quad (26)$$

From this equation the piston velocity corresponding to any crank position may easily be found. When the crank is in the top dead-center position ($\theta = 0°$), and in the bottom dead-center position ($\theta = 180°$), and $\sin\theta = 0$ the speed v of the piston is *nil.* Starting from the top dead-center position the speed of the piston increases until it reaches a maximum a little before the crank comes to the quarter position. The crank position corresponding to maximum piston speed may be found by placing the first differential coefficient of the expression in parentheses in equation (26) equal to zero. This gives

$$\cos\theta + \frac{1}{2n}(\cos^2\theta - \sin^2\theta) = 0;$$

$$2n\cos\theta = \sin^2\theta - \cos^2\theta;$$

$$\frac{2n}{\cos\theta} = \tan^2\theta - 1;$$

$$2n = \cos\theta(\tan^2\theta - 1).$$

Owing to the expedient resorted to when differentiating the expression for the momentary position of the piston, this last equation is not absolutely correct. The maximum speed position corresponds with the position for which

$$\tan\theta = 2n,$$

that is, when the crank arm and connecting rod make a right angle. The parts are shown in this position in Fig. 13,

$$\tan\theta = \frac{nl}{\frac{l}{2}} = 2n.$$

The variation of the piston speed during the course of one revolution of the crankshaft is shown graphically in Fig. 14, for an engine of 5-inch stroke with a 10-inch connecting rod and running 1800 r.p.m.

Piston Acceleration—To obtain an expression for the piston acceleration, we again differentiate equation (26), but before doing this we can simplify this equation by substituting $(\sin 2\,\theta)/2$ for its equivalent, $\sin\theta\cos\theta$. This gives us

$$v = \frac{\pi l N}{720}\left(\sin\theta + \frac{1}{4n}\sin 2\theta\right) \text{ ft. per sec.} \quad . \quad . \quad (27)$$

Then

$$\frac{dv}{dt} = \frac{\pi l N}{720}\left(\cos\theta + \frac{1}{2n}\cos 2\theta\right)\frac{d\theta}{dt}.$$

As before, $\frac{d\theta}{dt}$ is equal to $\frac{2\pi N}{60} = \frac{\pi N}{30}$, and, substituting, we have

$$a = \frac{\pi^2 N^2 l}{21{,}600}\left(\cos\theta + \frac{1}{2n}\cos\theta\right) \text{ ft. per sec. per sec.} \quad . \quad (28)$$

The acceleration is a maximum when $\theta = 0$, as $\cos\theta$ and $\cos 2\theta$ both reach their maximum value of 1. This is the dead-center position. Starting from the top dead-center the acceleration decreases and it becomes *nil* at the moment the piston reaches its maximum speed. It is, of course, self-evident that as the speed ceases to increase and is about to begin to decrease there must be a moment of no acceleration.

From the general expression for the force of acceleration,

$$F = \frac{Wa}{g},$$

we may, by inserting the value of the acceleration found in (28), derive the following formula for the force of acceleration of engine reciprocating parts

$$F_a = \frac{W}{g}\,\frac{\pi^2 N^2 l}{21{,}600}\left(\cos\theta + \frac{1}{2n}\cos 2\theta\right) \text{ lbs.},$$

which can be simplified to

$$F_a = \frac{WlN^2}{70{,}480}\left(\cos\theta + \frac{1}{2n}\cos 2\theta\right) \text{ lbs.}, \quad . \quad . \quad . \quad (29)$$

where W is the weight of the reciprocating parts in pounds, l the length of stroke in inches, N the number of revolutions per minute and n the ratio of connecting rod "center to center" length to length of stroke.

Weight of Reciprocating Parts—The inertia force is opposed to the force of explosion and materially reduces the shock received by the connecting rod and crank when the explosion takes place, when the engine is running at considerable speeds. The reciprocating weight W includes the weight of the piston and rings, the piston pin and one-third to one-half of the weight of the connecting rod. If an engine is to

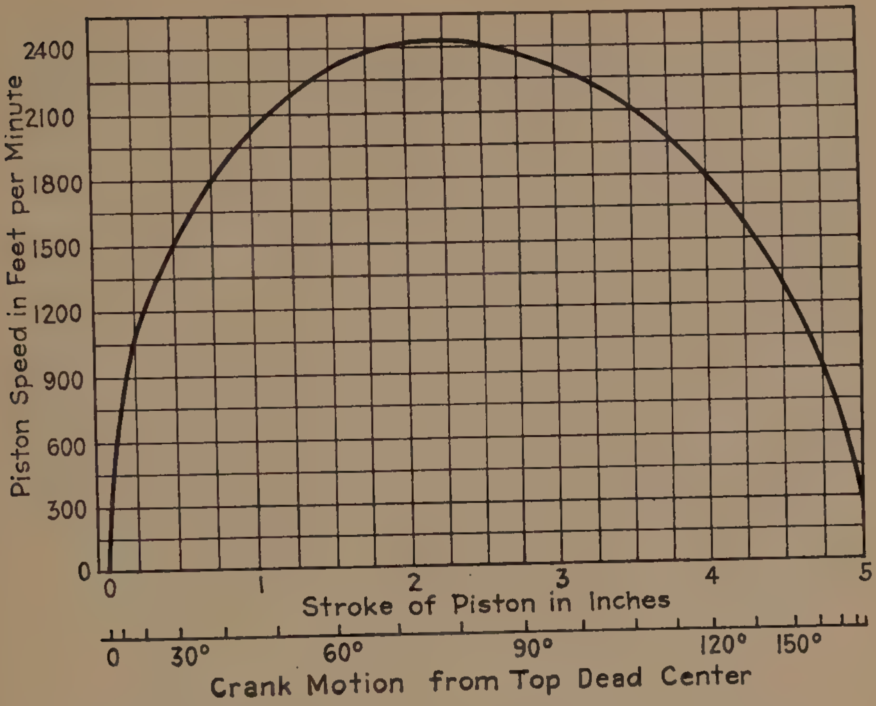


FIG. 14.—PISTON SPEED PLOTTED AGAINST PISTON MOTION.

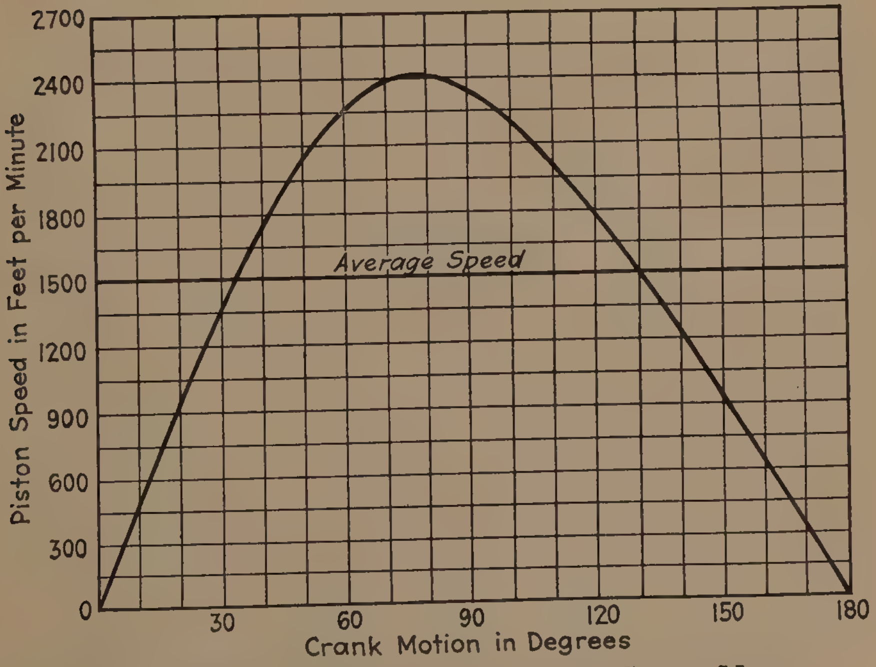


FIG. 15.—PISTON SPEED PLOTTED AGAINST CRANK MOTION.

be operated at high speed these parts must be made very light. They are made much lighter in automobile than in slow running stationary engines, and still lighter in racing automobile engines.

It is convenient to express the weight of the reciprocating parts in terms of the piston area, since the force of acceleration acts in opposition to the expanding gas during the first part of the power stroke and adds to it during the last part, and the force of expansion, as obtained from the indicator or manograph diagram, is expressed in pounds per square inch.

But while we may express the weight of the reciprocating parts in terms of piston head area, which varies as the square of the bore or as the square of the linear dimensions of the cylinder, we must not lose sight of the fact that this weight will increase approximately in direct proportion to the cube of the linear dimensions or to the piston displacement. As regards the fraction of the connecting rod weight which should be considered as reciprocating, it is obvious that the upper hub or bearing is entirely reciprocating weight, the lower hub or big end entirely rotating weight, and if the shank were of uniform section throughout it would be rational to consider half of its weight as reciprocating and the other half as rotating weight.

Owing to the relatively much greater diameter of the crankpin, as compared with the piston pin, especially in high speed, multi-cylinder engines, and to the fact that the shank generally tapers from the lower to the upper end, or at least has large fillets at the junction with the head, it is more logical in most cases to consider one-third of the connecting rod weight as reciprocating weight.

It appears from data published that the lightest pistons are not those made of the light alloys (with the possible exception of magnesium, which is not as yet much used in the United States), but iron pistons cast in permanent molds. With such pistons and connecting rods of forgeable light alloy, the total reciprocating weight may be as low as 0.22 lb. per sq. in. of piston head area, counting one-third of the connecting rod weight. With light iron pistons cast in sand, and steel connecting rods, the reciprocating weight will be about 0.32 lb. per sq. in. With aluminum pistons and steel connecting rods the figure for the reciprocating weight is somewhere between those given. In heavy truck engines, where no particular effort is made to lighten the reciprocating parts, the weight will range around 0.65 lb. per sq. in. of piston head area.

In the following, in applying the foregoing equations to a specific example, a type of engine is used whose characteristics are intermediate between those of the high-speed passenger car and the low-speed truck type.

The gas-pressure curve represented in Fig. 16 is a portion of a manograph diagram and shows a maximum pressure of 300 lbs. per sq. in. Now, let us consider an engine of 5-in. stroke whose reciprocating parts weigh 0.45 lb. per sq. in. It is required to find the force of inertia at different points of the stroke and the resultant pressure on the connecting rod.

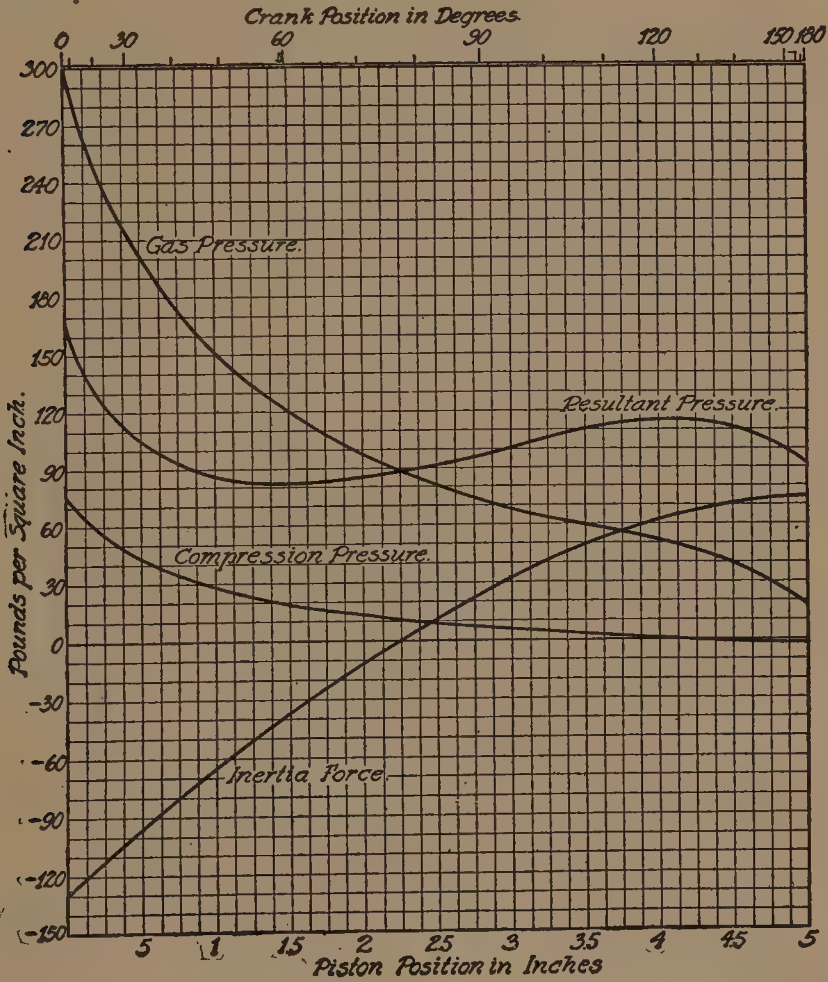


FIG. 16.—GAS PRESSURE, INERTIA AND RESULTANT CURVES FOR 3½ BY 5-IN. CYLINDER AT 1800 R.P.M. (RECIPROCATING WEIGHT, 0.45 LB. PER SQ. IN.)

It is first necessary to find the distance the piston has traveled from the top end of the stroke when the crank has turned through different angles from 0° to 180°. This can be done either by the construction of a diagram—in other words, graphically—or the values can be calculated by means of equation (24). The values thus obtained are given in the second column of Table II. The piston speeds corresponding to different crank angles are next determined by means of equation (27). Then the piston acceleration is determined by means of equation (28), and finally the force of acceleration by means of equation (29). All of the values obtained are given in Table II. The inertia force curve is now drawn in in Fig. 16.

Table II—Acceleration Forces in an Engine of 3.5-Inch Bore, 5-Inch Stroke, Connecting Rod Length Twice the Stroke and Reciprocating Parts Weighing 0.45 Pound Per Square Inch of Piston Head Area at 1800 R.P.M.

Degrees	Piston Position (Inches)	Piston Speed (Feet per Min.)	Piston Acceleration (Feet per Sec. per Sec.)	Force of Acceleration Pounds per Square Inch)
0	0	0	9247	129.5
10	0.048	518	9027	126.3
20	0.190	995	8179	114.3
30	0.415	1433	7166	100.2
40	0.720	1762	5854	81.8
50	1.047	2094	4435	62.0
60	1.485	2295	2711	37.9
70	1.925	2397	1114	15.6
80	2.375	2421	−450	−6.3
90	2.815	2355	−1881	−26.3
100	3.245	2220	−3026	−42.4
110	3.635	2020	−3944	−55.2
120	3.985	1785	−4624	−64.7
130	4.292	1514	−5078	−72.4
140	4.550	1224	−5346	−74.7
150	4.745	922	−5409	−77.5
160	4.890	616	−5542	−77.6
170	4.972	307	−5548	−77.6
180	5.000	0	−5548	−77.6

At the beginning of the stroke the reciprocating masses must, of course, be set in motion, and they then consume some of the energy of the expanding charge. When the crank has turned through an angle of about 76°, the piston reaches its maximum speed; thereafter its speed decreases and it returns the energy which it absorbed during the first part of the stroke. By combining the gas pressure curve and the inertia force curve we obtain the curve of the resultant pressure, as shown in Fig. 16. It will be noticed that the inertia of the reciprocating parts greatly reduces the maximum pressure on the connecting rod.

In the same way that we plotted in Fig. 16 the resultant pressure in the direction of piston travel for the power stroke, we may plot this pressure for the other strokes. In doing this the gaseous pressures during the admission and exhaust strokes may be neglected, as they are exceedingly small in comparison with the inertia forces at normal speed. The pressure of the gas during compression must, however, be taken into account. All forces which oppose the motion of the piston are considered negative and are plotted from the base line downward, while all forces which assist the piston in its motion are considered positive and are plotted from the base line up. In Fig. 17 is shown such a diagram for a complete cycle, the assumptions being the same as for Fig. 16. Starting with the admission stroke, the only force acting is that of inertia, and since this is a down stroke, the inertia is exactly the same as for the power stroke, as plotted in Fig. 16. Exigencies of space made it necessary to use in Fig. 17 a horizontal scale half as large as in Fig. 16.

Next comes the compression stroke, during which both gas pressure and inertia forces are acting. Owing to the fact that this is an up stroke, the inertia force diagram is the reverse of the inertia force diagram for the previous stroke. As for every other stroke, the inertia force is at first negative and toward the end positive. The gas pressure during the compression stroke is negative throughout, and in the diagram is represented by a dash-dotted line, while the inertia force for this stroke is represented by a dotted line. The resultant of the two forces is represented by a full line.

The next following portion of the diagram is an exact reproduction of Fig. 16, except that the horizontal scale is only half as large. During the last stroke, while the cylinder is exhausting, only the inertia force is acting, and this is again the same as during the compression stroke. The curve shown in a heavy full line represents, therefore, the variations of the effective pressure in the direction of piston travel throughout the four strokes of the cycle.

In the construction of the inertia curves it is convenient to make use of a table of "crank angle factors." The expression for the inertia force, viz.:

$$\frac{WlN^2}{70{,}480}\left(\cos\theta + \frac{1}{2n}\cos 2\theta\right) \text{ lbs.},$$

may be considered as composed of two parts, one including the different values of the particular engine and the constant, and the other depending entirely upon the position of the crank, and upon the ratio of the connecting rod center to center

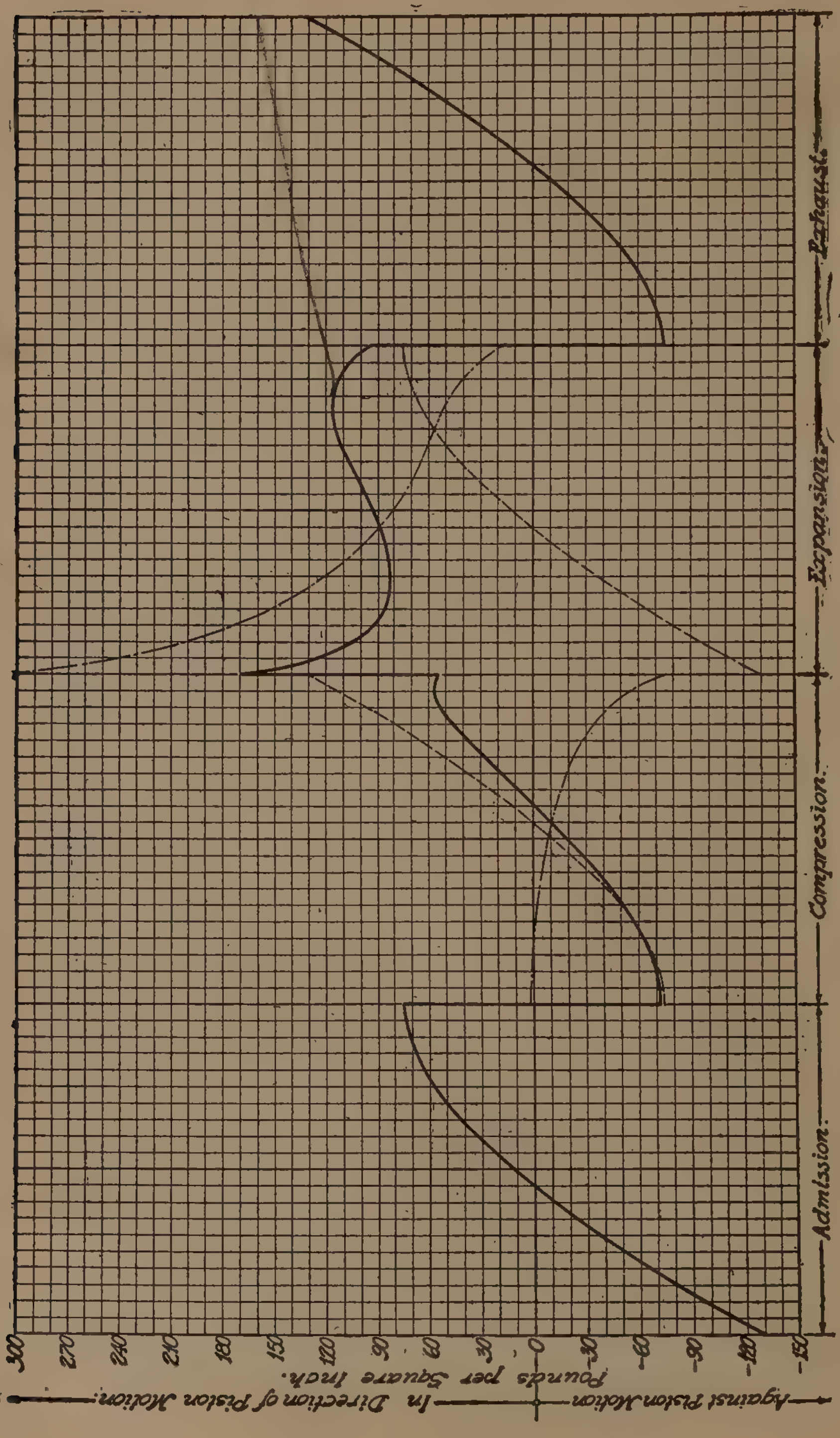


FIG. 17.—DIAGRAM OF RESULTANT PRESSURE ALONG THE AXIS OF A 3.5 BY 5-IN. CYLINDER.

length to the length of stroke. This latter factor ($\cos \theta + \cos 2\theta/2n$) we will call the "crank angle factor," and its values for angles from zero to 180° in increments of 10° and for values of the ratio n from 1¾ to 2½, inclusive, in increments of ⅛, are given in Table III. A rather peculiar feature in connection with the crank angle factor is that for values of n above 2 the factor is largest for 0° and 180° and smaller for all intermediate angles, while for values of n below 2 the factor decreases again as the crank angle approaches 180°; in other words, the inertia force reaches its second maximum some time before the lower dead center is reached.

Crank Moment—From Fig. 17 it can be seen that the pressure in the direction of piston travel is a maximum at the beginning of the power stroke. The turning effort on the crank at that moment is *nil,* however, for the reason that the connecting rod and crank arms are then in line with each other, and all the pressure is spent in producing thrust on the crankshaft bearings.

The turning moment is the product of a force into the perpendicular distance from the line in which the force acts, to the center of rotation, and is expressed in pound-inches or pound-feet.

Table III.—Crank Angle Factor of Inertia Force

	$n =$						
	1¾	1⅞	2	2⅛	2¼	2⅜	2½
0°	1.286	1.237	1.250	1.235	1.222	1.211	1.20
10°	1.254	1.236	1.220	1.206	1.194	1.183	1.173
20°	1.158	1.144	1.131	1.120	1.110	1.101	1.093
30°	1.009	0.999	0.991	0.984	0.977	0.971	0.966
40°	0.816	0.812	0.809	0.807	0.805	0.803	0.801
50°	0.593	0.596	0.599	0.602	0.604	0.606	0.608
60°	0.357	0.367	0.375	0.382	0.389	0.395	0.400
70°	0.124	0.138	0.151	0.162	0.172	0.181	0.189
80°	−.095	−.077	−.061	−.047	−.035	−.024	−.014
90°	−.286	−.266	−.250	−.235	−.222	−.210	−.200
100°	−.422	−.425	−.409	−.395	−.383	−.372	−.362
110°	−.560	−.546	−.533	−.522	−.512	−.503	−.495
120°	−.643	−.633	−.625	−.618	−.611	−.605	−.600
130°	−.693	−.689	−.686	−.684	−.682	−.680	−.678
140°	−.716	−.720	−.723	−.725	−.727	−.729	−.731
150°	−.723	−.733	−.741	−.748	−.755	−.761	−.766
160°	−.722	−.736	−.749	−.760	−.770	−.779	−.787
170°	−.717	−.734	−.750	−.764	−.776	−.787	−.797
180°	−.714	−.734	−.750	−.765	−.778	−.790	−.800

The pressure in the direction of piston travel is held in balance by two reactions, viz., the reaction of the connecting rod on the piston pin and the reaction to the side thrust of the piston against the cylinder wall. Fig. 18 is a so-called parallelogram of forces. If we denote the pressure in the direction of piston travel by P_p, the pressure acting along the connecting rod by P_c and the side thrust reaction of the cylinder wall by P_s, then, obviously,

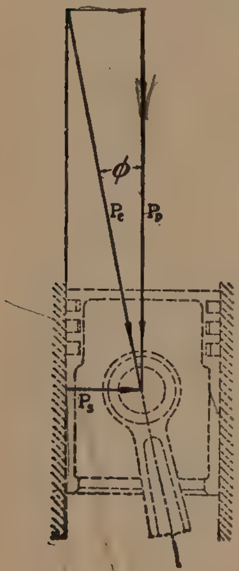


FIG. 18.

$$P_c = \frac{P_p}{\cos \phi}, \qquad (30)$$

and

$$P_s = P_p \tan \phi. \qquad (31)$$

In Figs. 19 and 20 are shown two diagrams of the crank and connecting rod with the crank at an angle θ beyond the top dead center, which angle is less than 76° in Fig. 19 and more than 76° in Fig. 20. The lever arm through which the force F_c, acting along the connecting rod, attacks the crank is

$$OC = \frac{l}{2} \cos \alpha.$$

But

$$\alpha = 90° - (\theta + \phi),$$

in Fig. 19 and

$$\alpha = -[90° - (\theta + \phi)],$$

in Fig. 20, hence, generally,

$$\alpha = \pm [90° - (\theta + \phi)],$$

and, since the cosine of an angle is equal to the sine of its complement,

$$\cos \alpha = \pm \sin (\theta + \phi), \qquad (32)$$

and

$$\frac{l}{2} \cos \alpha = \pm \frac{l}{2} \sin (\theta + \phi),$$

which may be written in the form

$$\pm \frac{l}{2} (\sin \theta \cos \phi + \cos \theta \sin \phi).$$

Multiplying the force by the lever arm through which it acts, we get for the turning moment

$$M = \frac{P_p}{\cos\phi} \times \frac{l}{2}(\sin\theta\cos\phi + \cos\theta\sin\phi)$$

$$= \frac{P_p l}{2}\left(\sin\theta + \cos\theta\frac{\sin\phi}{\cos\phi}\right). \quad \ldots\ldots \quad (33)$$

Referring again to Fig. 19,

$$\sin\phi = \frac{\sin\theta}{2n},$$

and

$$\cos\phi = \sqrt{1 - \frac{\sin^2\theta}{4n^2}} = 1 - \frac{\sin^2\theta}{8n^2}$$

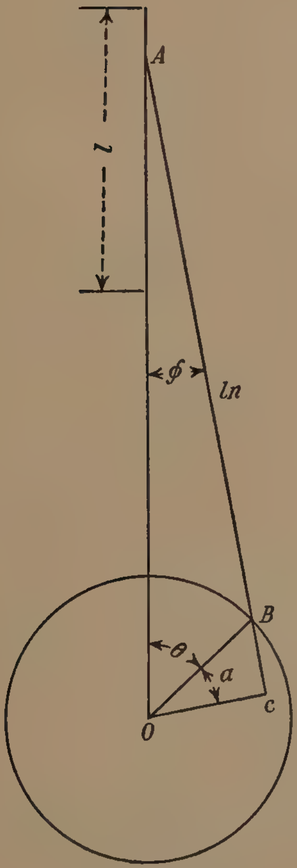


FIG. 19.

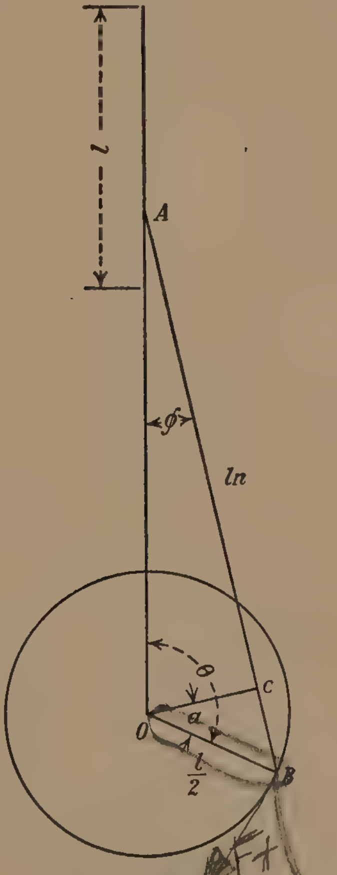


FIG. 20.

(approximately). Consequently,

$$\frac{\sin\phi}{\cos\phi}=\frac{\dfrac{\sin\theta}{2n}}{1-\dfrac{\sin^2\theta}{8n^2}}=\frac{4n\sin\theta}{8n^2-\sin^2\theta}.$$

Substituting in (33) we get

$$M=\frac{P_pl}{2}\left(\sin\theta+\frac{4n\sin\theta\cos\theta}{8n^2-\sin^2\theta}\right).\quad .\quad .\quad .\quad (34)$$

This equation gives the turning moment per square inch of piston head area in pound-inches. It is usually desired, however, to get the total turning moment expressed in pound-feet, and to this end the value for M in equation (34) has to be multiplied by the piston head area and divided by 12. This gives for the total crank turning moment

$$M_t=\frac{\pi b^2P_pl}{96}\left(\sin\theta+\frac{4n\sin\theta\cos\theta}{8n^2-\sin^2\theta}\right)\ \textit{pound-feet},.\quad (35)$$

which equation enables us to determine the value of the turning moment on the crank for any angular position. In this equation $\sin^2\theta$ is quite small as compared with $8n^2$, and it may be neglected in approximate calculations, in which case the equation can be reduced to the simple form

f.
ax. turning
load

$$M_t=\frac{\pi b^2lP_p}{96}\left(\sin\theta+\frac{\sin\theta\cos\theta}{2n}\right)\ \textit{pound-feet}.$$

The moment is *nil* for $\theta=0°$ and $180°$, as in both of these cases $\sin\theta=0$. It reaches a maximum toward the end of the first quarter revolution. For $\theta=90$ degrees, $\cos\theta=0$ and the second part of the term in parentheses vanishes. The turning moment on the crank is then equal to the product of the pressure on the piston into the length of the crank arm—

$$M=\frac{\pi b^2lP_p}{96}\ \textit{pound-feet}.$$

If we divide the turning moment by the length of the crank-arm we get the tangential effort F_t. Hence

$$F_t=\frac{\pi b^2P_p}{4}\left(\sin\theta+\frac{4n\sin\theta\cos\theta}{8n^2-\sin^2\theta}\right)\ \textit{pounds}.\quad .\quad (36)$$

For $\theta=90°$ we have

$$F_t=\frac{\pi b^2P_p}{4}\ \textit{pounds}.$$

The motion of the crank pin is then for a moment parallel to the motion of the piston and since, owing to the connecting rod connection, their distance apart cannot change, they must travel at the same speed, and the pressures on them in the vertical direction must be the same.

In Fig. 21 is shown a crank moment diagram for a single cylinder engine of 3½ in. bore, 5-in. stroke, 10-in. connecting rod length, 0.45 pound reciprocating weight per square inch of piston head area and indicating a compression pressure of about 70 lbs. per sq. in. gauge and an explosion pressure of 300 lbs. per sq. in. gauge.

In drawing a diagram of this kind it is convenient to make use of a table of crank angle factors for crank pin turning moment, like Table IV. This table gives the values of the expression

$$\sin\theta + \frac{4n\sin\theta\cos\theta}{8n^2 - \sin^2\theta}.$$

of equation (35). From a diagram similar to Fig. 17, we obtain the piston pressure P_p for any crank angle. This we multiply by the area of the piston head

$$\frac{3.5 \times 3.5 \times 3.1416}{4} = 9.62 \text{ sq. in.}$$

Table IV.—Crank Angle Factor of Crank Moment

	$n = \frac{\text{Connecting rod}}{\text{stroke}}$						
θ	$1\frac{3}{4}$	$1\frac{7}{8}$	2	$2\frac{1}{8}$	$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{1}{2}$
0°	0.	0.	0.	0.	0.	0.	0.
10°	0.223	0.220	0.217	0.214	0.212	0.210	0.208
20°	0.435	0.428	0.423	0.418	0.414	0.410	0.407
30°	0.626	0.617	0.610	0.603	0.598	0.592	0.588
40°	0.788	0.778	0.769	0.761	0.752	0.748	0.743
50°	0.913	0.900	0.894	0.886	0.879	0.873	0.868
60°	1.177	0.988	0.980	0.973	0.966	0.957	0.955
70°	1.038	1.031	1.025	1.019	1.015	1.011	1.007
80°	1.037	1.033	1.030	1.028	1.025	1.023	1.021
90°	1.	1.	1.	1.	1.	1.	1.
100°	0.933	0.936	0.939	0.942	0.945	0.947	0.949
110°	0.842	0.849	0.857	0.860	0.865	0.869	0.873
120°	0.735	0.745	0.753	0.759	0.766	0.771	0.776
130°	0.624	0.629	0.638	0.646	0.653	0.659	0.665
140°	0.497	0.508	0.517	0.524	0.531	0.537	0.542
150°	0.377	0.383	0.390	0.397	0.402	0.408	0.413
160°	0.250	0.253	0.261	0.266	0.270	0.274	0.277
170°	0.125	0.127	0.130	0.133	0.135	0.137	0.139
180°	0.	0.	0.	0.	0.	0.	0.

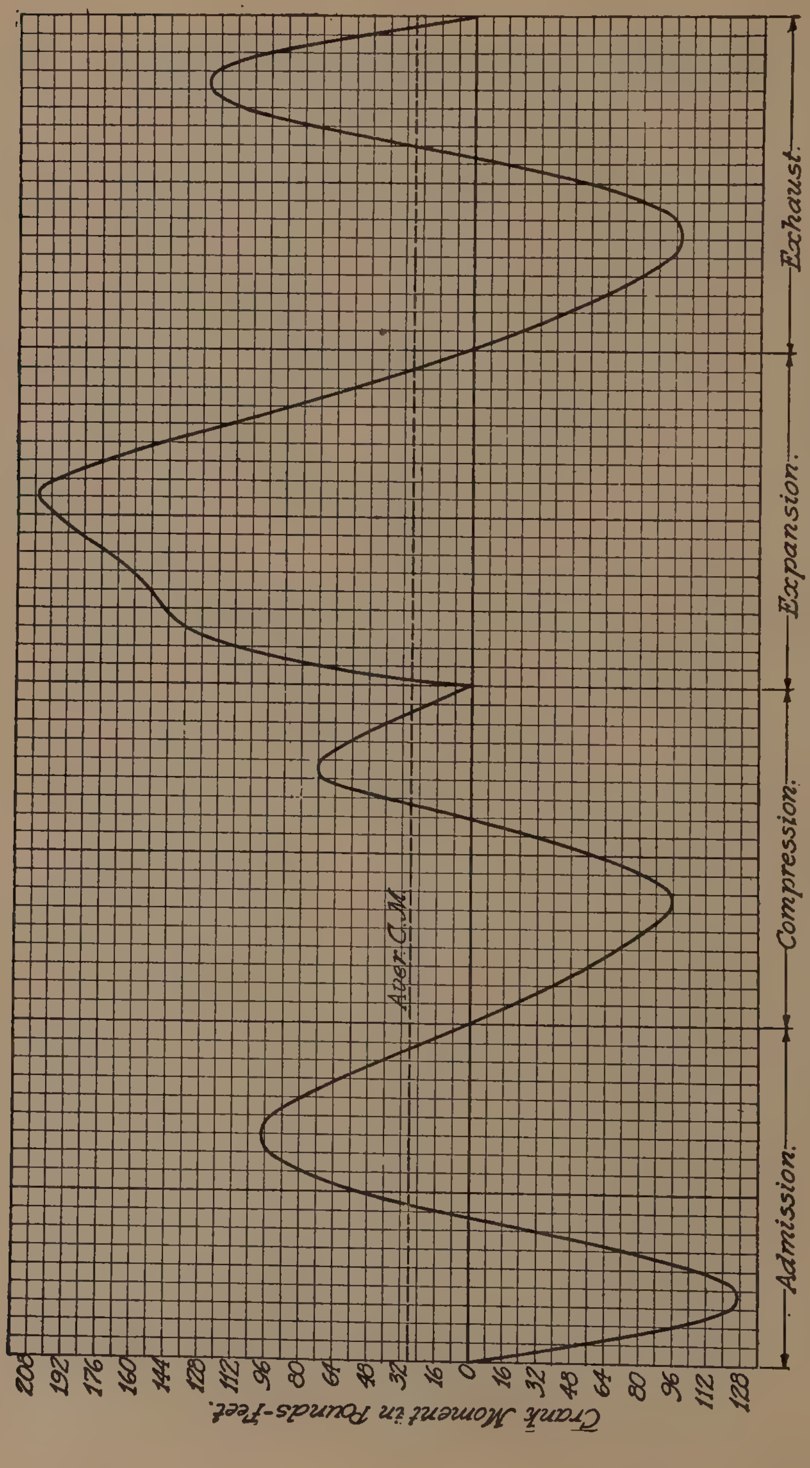


FIG. 21.—TURNING MOMENT DIAGRAM OF SINGLE CYLINDER ENGINE.

and then multiply by the value of $l/2$, viz., 2.5 inches, and divide the result by 12 in order to obtain the final result in pound-feet. The result obtained is multiplied by the crank angle factor obtained from Table IV, and the values thus arrived at are plotted as ordinates in a diagram (Fig. 21) with the crank positions as the abscissas. It will be observed that, starting from the beginning of the admission stroke, the crank turning moment is negative until the crank has turned through about 76°; then it becomes positive and remains so until the end of the stroke. During the first part of the compression stroke for about 107° of crank motion the turning moment is negative again; then it becomes positive, and just before the end of the compression stroke, as the compression pressure overpowers the inertia force, it becomes momentarily negative again. During the whole of the power stroke the turning moment is positive. At about the middle of the power stroke there is a peculiar dip in the curve. This is due to the fact that as the crank approaches the 90° position there is a considerable increase in the piston pressure, owing to the reversal of the inertia force, while the gas pressure and crank angle are nearly constant. During the last stroke, while the engine is exhausting, only inertia forces are acting, and the turning moment is negative during the first part and positive during the last part.

Figure 22 is a crank moment diagram for a two-cylinder engine of either the opposed type or of the twin type with both pistons working on a single crank pin. In both of these engine types the explosions occur at equal intervals of one revolution, and both of the pistons start simultaneously on corresponding strokes. The curve of Fig. 22 is based on the assumption that both cylinders are of 3.5-inch bore and 5-inch stroke, and the other assumptions are the same as in the previous example. In such two cylinder engines the power strokes are separated by only a single idle stroke. It will be seen that the maximum turning moment is only a little greater than in the case of a single cylinder engine, although the power and the average turning moment are twice as great. The maximum negative turning moment is, however, also about twice as great in the two cylinder as in the single cylinder engine. The best criterion of the relative uniformity of turning moment is undoubtedly the ratio of the total turning moment range to the average turning moment. This is about 8.5 in the case of the single cylinder, and about 6.8 in the case of the double cylinder engine.

Figure 23 is the turning moment diagram of a four-cylinder engine of 3.5 in. bore and 5 in. stroke. Under the

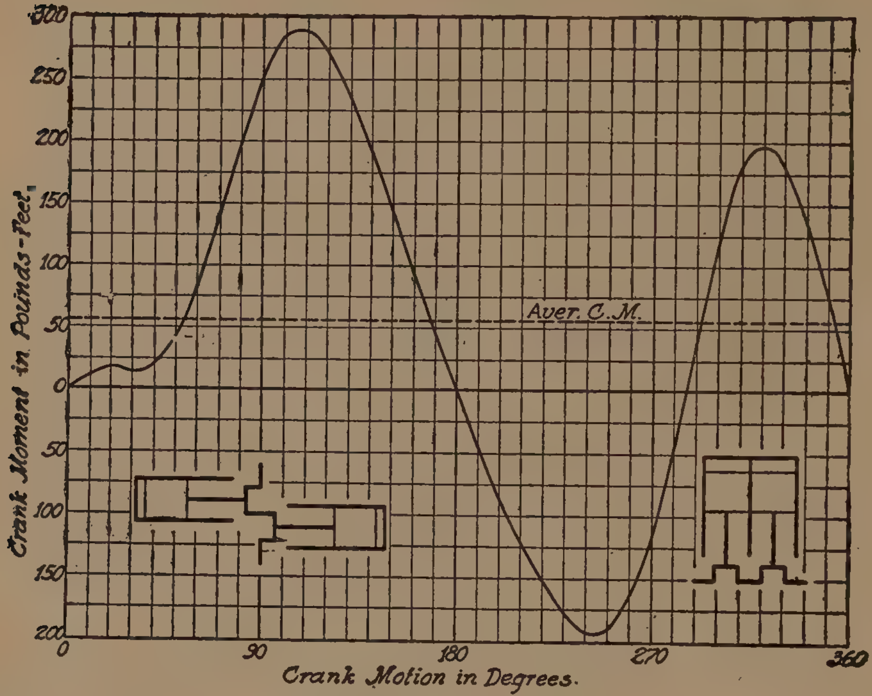


FIG. 22.—CRANK MOMENT DIAGRAM OF TWO-CYLINDER ENGINE.

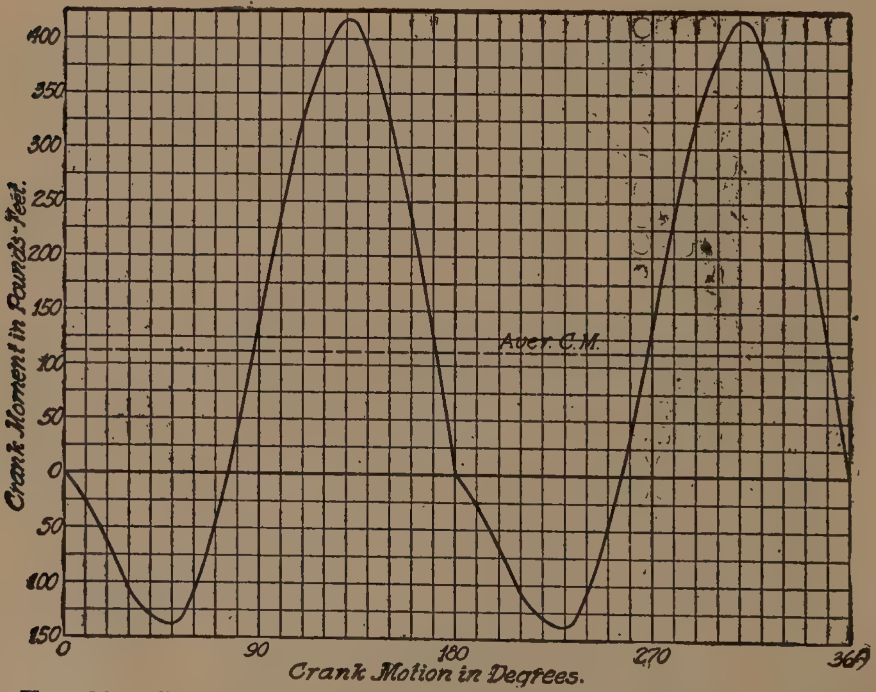


FIG. 23.—CRANK MOMENT DIAGRAM OF FOUR-CYLINDER ENGINE.

assumptions here made the turning moment reverses in direction at the beginning of each stroke and also 76° farther on. With light reciprocating parts or at lower speed there may be no reversal of the turning moment, though the latter must necessarily always be *nil* at the beginning of a stroke. The ratio of the total turning moment range to the average turning moment is 5. This diagram very strikingly shows the effect of the inertia of the reciprocating parts. Although the gas pressure in the cylinder is far greater during the first half of the down stroke than during the last half, the turning moment on the crank is negative during the first half of the stroke, and attains its maximum value during the last half, at about 130° from the top dead center.

In Fig. 24 are shown turning moment diagrams of six-eight- and twelve-cylinder, 288 cubic-inch engines. In the case of the six-cylinder, the ratio of the maximum turning moment range to the average turning moment is 1.63. Of course, these different factors of turning moment fluctuation, viz., 8.5 for the single cylinder, 6.8 for the double-cylinder, 5 for the four-cylinder, and 1.63 for the six-cylinder, apply strictly only if the assumptions here made regarding gas pressure variation, weight of reciprocating parts and relative length of connecting rod hold true, but the factors would be very nearly the same for any practical automobile engine, and the above figures may well be taken as indices of the relative uniformity of turning moment in engines with various numbers of cylinders. In the eight-cylinder type engine the ratio of maximum variation to average turning moment is about 0.58, thus showing a further improvement over the six-cylinder engine. The eight-cylinder compares even more favorably with the six in this respect when low-speed turning moments are considered. Finally, in the twelve-cylinder or twin six engine the ratio of maximum variation to average turning moment is only 0.2 and the torque is nearly constant.

The crank moment diagrams may also be made to serve as crank effort diagrams by suitably dividing the vertical scale. Since the crank arm in all the foregoing examples is 2½ inches = 0.208 ft., the tangential effort in pounds on the crank pin for any angular position of the crank is equal to the corresponding crank moment in pound-feet divided by 0.208. However, to get the full pressure on the crank pin it is necessary to combine the effort thus found with the centrifugal force on the connecting rod head and lower part of the shank, which can be done graphically.

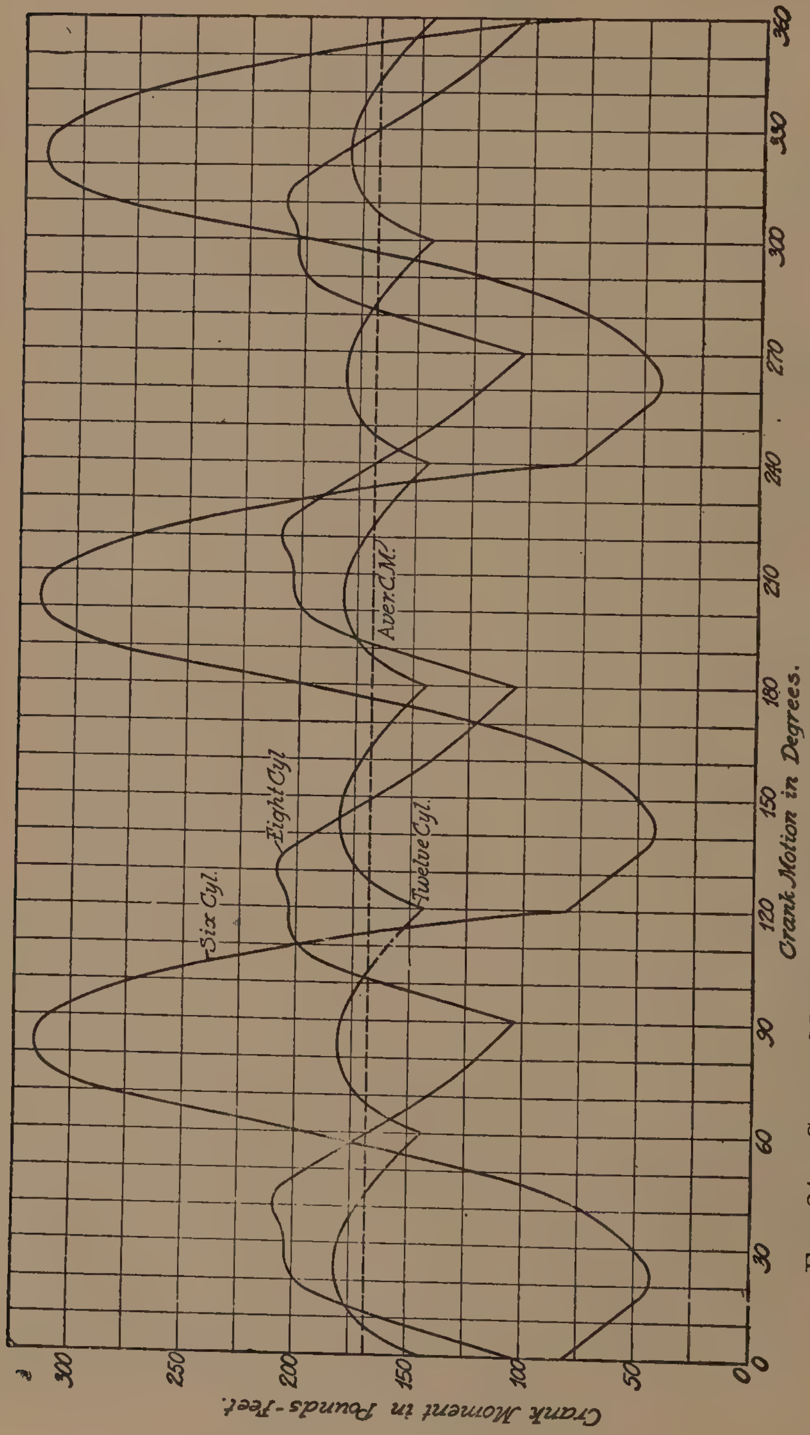


Fig. 24.—Crank Moment Diagrams of Six-, Eight- and Twelve-Cylinder Engines.

CHAPTER V

Balancing of Engines

There are four possible causes of vibration in gasoline engines, viz.:

(1) Unbalanced rotating parts.
(2) Unbalanced reciprocating parts.
(3) Torque reaction.
(4) Torsional flexibility of the crankshaft.

Aside from the flywheel, which is a body of revolution and will be balanced if accurately made and free from blow-holes and other irregularities, the only rotating part of considerable inertia is the crankshaft. If an ordinary single throw crankshaft is rotated at considerable speed, the centrifugal force acting on its unbalanced members, that is, the crank arms and the crank pin, is transmitted to the engine frame and always tends to displace the engine in the direction in which the crank arms are pointing at the moment. Since the crank arms are rotating uniformly, the general effect, if the engine were absolutely free, would be to displace its crankshaft center line in a circle. The rigid connection of the engine to the chassis frame, as well as the more or less flexible connection of the latter to the axles and wheels, materially modifies and reduces the amplitude of the resulting vibration, but does not eliminate it entirely.

Crankshaft Balance—To obviate such vibration the crankshaft must be balanced; that is, it must be provided with counter-weights extending from it radially in the opposite direction from the crank arms, so proportioned that if the crank were placed on a pair of steel balance ways it would remain in any position in which it were placed. (See Fig. 25.) The general principle of rotating balance may be stated as follows:

If we cut a thin section from the rotating body, perpendicular to its axis of rotation and at any point along its

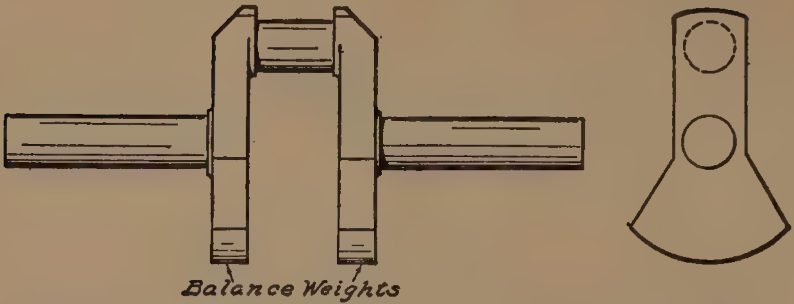


FIG. 25.—SINGLE THROW CRANK WITH BALANCE WEIGHTS.

length, and then draw any line across the section and through the axis, the sum of the products of all particles of the section to one side of this line, into their respective distances from the axis (their moments) must be equal to the sum of the products of all particles on the other side of the line into their respective distances from the axis.

From the above it is obvious that a crankshaft cannot be perfectly balanced, as it is impossible to place any weight opposite the crankpin, or at least, opposite its central section. However, owing to the comparatively high rigidity of the crankshaft, substantially the same result as from perfect balance is obtained by placing the weight for balancing the crank pin opposite the crank arms.

If a crankshaft were absolutely rigid, then a multiple-throw crankshaft with the throws symmetrically distributed around the circle and with each throw duplicated by another in the same radial plane and at the same distance from the center of the crankshaft but in the opposite direction, would be inherently balanced. However, a crankshaft is not perfectly rigid and the centrifugal force on any throw creates a load on the bearings on opposite sides of that throw even when the crankshaft has another, oppositely located throw. Besides, since the two centrifugal forces act on the crankshaft along parallel lines some distance apart, and in opposite directions, they create a rotating couple.

The balancing of the rotating parts presents no particular difficulty in any type of engine, as an equivalent balancing mass can be placed at least nearly opposite each such part. The reciprocating parts, however, are not so easy to deal with. Considering first the case of the single cylinder engine, its reciprocating parts might be balanced by outside weights connected to the crankshaft in such a manner that they would be moved at every instant at the same speed as the engine piston but in the opposite direction. There should be two such weights, one on either side of the engine, each equal to half the weight of the reciprocating parts. Such an arrangement is not practical, however, because instead of complicating a single cylinder engine in this way, any designer would rather adopt a multicylinder type.

Reciprocating Masses—It is, however, possible to partly balance a reciprocating mass by a revolving mass. This may be explained as follows: A revolving mass is acted upon by a radial force (centrifugal force)

$$F = \frac{WN^2r}{35{,}240}\text{ lbs.}, \quad \ldots \ldots \ldots \quad (37)$$

where W is the weight in pounds, N the number of revolutions per minute and r the radius in inches. Such a rotating force can be resolved into two periodic forces at right angles to each other. We will resolve it into a vertical and a horizontal component. Referring to Fig. 26, let OA represent the force in magnitude and momentary direction. Then the horizontal component

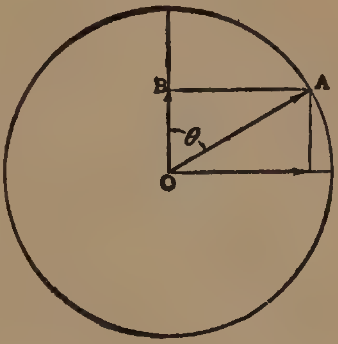


FIG. 26.

$$BA = OA \sin \theta$$

and the vertical component

$$OB = OA \cos \theta = OA \sin (\theta + 90°).$$

When $\theta = 0$, that is when the weight is directly above the center of rotation, the vertical component is equal to the total force and the horizontal component is *nil*. The same when the weight is directly below the center of rotation, when $\theta = 180°$. Each of the components can be represented by a sine curve; that is, a curve whose abscissas represent angles and whose ordinates are proportional to the sines of the angles represented by the corresponding abscissas.

Single-cylinder Engine—We found the expression for the inertia force due to the reciprocating parts to be (equation 29)

$$F_a = \frac{WlN^2}{70{,}480}\left(\cos \theta + \frac{l}{2n} \cos 2\theta\right).$$

It will be observed that the factor enclosed in parentheses is made up of two terms, viz., $\cos \theta$ and $(l/2n) \cos 2\theta$. Each of these terms when plotted forms a sine curve, as shown in Fig. 27. It will be seen that the second term is of twice the frequency as the first; that is, during a given time (number of crank revolutions) it passes through zero and through its maximum value twice as often as the first term. The first term in the parentheses multipled by the factors outside the parentheses, viz., $WlN^2 \cos \theta/70{,}480$, is known as the primary unbalanced force, and the second term multiplied by the factors outside the parentheses, viz., $WlN^2 \cos 2\theta/(2n \times 70{,}480)$, is known as the secondary unbalanced force. With an infinitely long connecting rod the latter would vanish.

In a single-cylinder engine the primary unbalanced force can be partly balanced (or be reduced in value) by means of rotating balance weights connected to the crank arms, as shown in Fig. 25. If in the expression for the primary unbalanced force we use the crank radius r instead of the length of stroke l, the value of the denominator in the fraction will be halved and the expression then becomes $WlN^2 \cos \theta/35{,}240$, so that the maximum value of the primary unbalanced force is $WlN^2/35{,}240$. But this is exactly the same expression as that for the centrifugal force on a rotating weight W with a radius of rotation r. Hence, if balance weights equal to the reciprocating weights were placed opposite the crank arms at an average distance from the axis of rotation equal to the crank radius, they would exactly neutralize the primary unbalanced force when the latter is a maximum, that is, in the dead center positions of the crank.

However, in the 90° positions the balance weights would create a horizontal force, and as the forces due to the reciprocating weights are at all times vertical (in a vertical engine), there would be nothing to balance this horizontal force. In fact, only the vertical component of the centrifugal force on the balance weights would be required to neutralize the primary unbalanced force due to the reciprocating parts, and the net result of the arrangement described would be to transform the vertical reciprocating force into an equal horizontal one, whereby nothing would be gained. But if we provide the crankshaft with such balance weights that the centrifugal force on them is equal to only one-half the primary inertia force on the reciprocating parts, then the latter will be reduced by one-half, and a horizontal component equal to it will be introduced. The maximum unbalanced force is therefore halved, and the amplitude of the vibration is also halved. In practice the balance weights are so designed that most of their mass is as far removed as possible from the axis of rotation, as it is not the weight but the moment of the weight that counts.

Torque Reaction—The third cause of vibration is the torque reaction. When the connecting rod impresses a certain turning moment upon the crankshaft, the side thrust of the piston against the cylinder wall causes an equal and opposite moment to be impressed upon the engine frame. This can easily be shown algebraically, but it is self-evident, since, according to the laws of mechanics, action and reaction are always equal and opposite. It was once believed that this cause of vibration could be eliminated by having two equal engines side by side, in a single block and turning synchronously in opposite directions by being geared together. This solution of the problem was repeatedly suggested in the early years of automobile history, but was never applied to any extent. It was thought preferable to use multicylinder engines, thus reducing the size of the individual cylinders and the power of the individual explosions. The fluctuations in the turning moment will then be much smaller. It must be remembered that what causes vibration is not an unbalanced force but the fluctuations of such a force. A steady uniform turning moment might twist the engine slightly out of its normal position, but it would remain in that position as long as the torque was maintained, and there would be no vibration. But if the moment varies periodically, or even changes in direction, then the engine will swing back and forth, or vibrate.

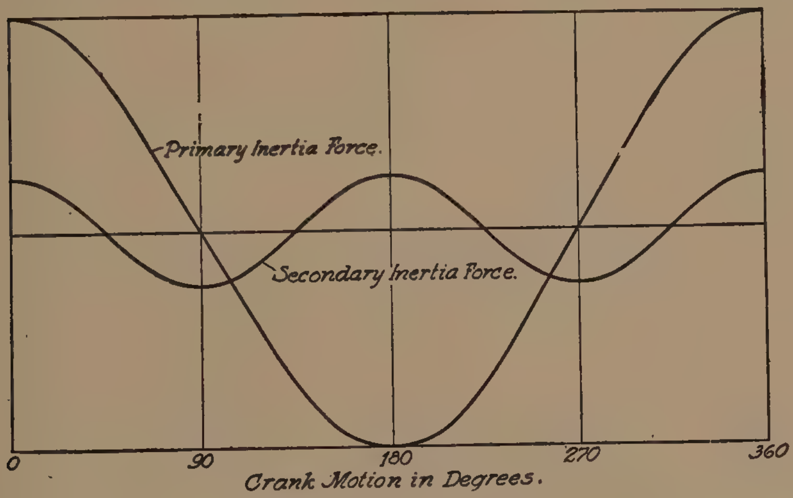


Fig. 27.—Primary and Secondary Harmonics of Inertia Force.

Two-Cylinder Engine—Considering next the two-cylinder engine, the only type that has been used to any extent for automobile work is that in which the cylinders are located horizontally on opposite sides of the crankshaft, the latter having double throws. Although such engines have been made with two cylinders exactly in line with each other, generally they are slightly displaced sidewise with relation to each other. Since the two pistons start simultaneously from the outer end of the stroke (which corresponds to the top end in a vertical engine) their velocities are always equal, and since the two sets of reciprocating parts are equal in weight, their inertia effects would neutralize each other if it were not for the slight sidewise displacement of the two cylinders. As it is, the forces form a couple in a horizontal plane, proportional to the distance between the cylinder center lines. The moving parts in such an engine are thus very nearly balanced, and since the range in the torque reaction for an engine of a given output is much smaller than in the single cylinder, especially at low speed, a double-cylinder opposed engine runs relatively free from vibration.

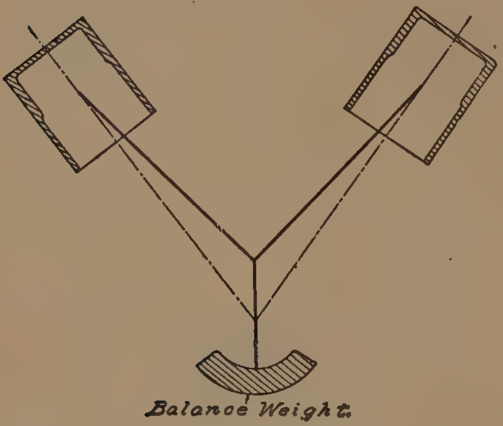


Fig. 28.—Diagram of 90° V Engine.

V-Type Engine—Two-cylinder engines with the cylinders set at an angle of 90° (and often less) are used to quite an extent on motorcycles. They are known as V-type engines and have a relatively good balance, considering the small number of cylinders.

We have in such an engine two sets of reciprocating parts moving at right angles to each other, so that if one set moved vertically up and down, the other would move horizontally back and forth. The primary unbalanced forces of the two sets of reciprocating masses, each of which is represented by a sine curve, combine to make a single constant radial rotating force which can be balanced by means of suitable balance weights placed on the crank arms opposite the crank pin, as shown in Fig. 28. Hence the primary unbalanced force is eliminated in the two-cylinder V engine, but the secondary unbalanced force, which has a frequency twice that of crankshaft rotation, remains.

Four-Cylinder Engine—In a four-cylinder engine the two inside pistons work on cranks in the same plane, and the two outside pistons on cranks directly opposite the former (Fig. 29). The two inside pistons then always move up and down together, and the two outside pistons at all times move in the opposite direction to the inside pistons. The center lines of motion of the two sets of reciprocating masses coincide. However, although the two sets of reciprocating masses are equal in weight and always move in opposite directions, the inertia forces do not completely neutralize each other, for the reason that the pistons travel faster during the upper than during the lower half of the stroke.

If we represent the momentary inertia force due to the reciprocating masses in one cylinder by

$$c\left(\cos\theta + \frac{1}{2n}\cos 2\theta\right),$$

where

$$c = WlN^2/70{,}480,$$

then the force due to any two cylinders whose reciprocating masses are in phase—say the two outside cylinders—is twice this, or

$$2c\left(\cos\theta + \frac{1}{2n}\cos 2\theta\right).$$

The inertia force due to the reciprocating masses of the two inner cylinders is

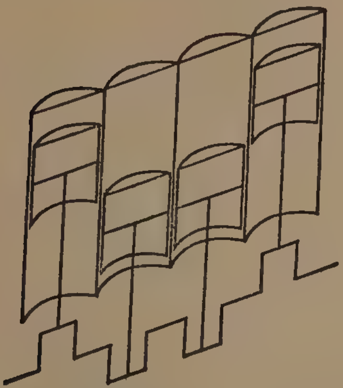


FIG. 29.—SKETCH OF FOUR-CYLINDER ENGINE.

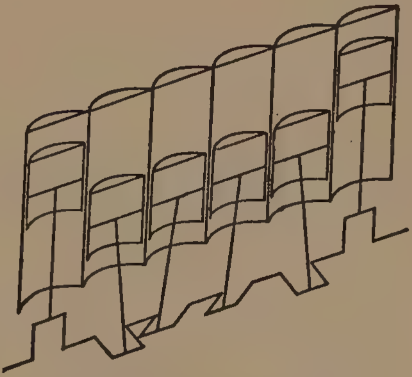


FIG. 30.—SKETCH OF SIX-CYLINDER ENGINE.

$$2c\left[\cos(\theta + 180^\circ) + \frac{1}{2n}\cos 2(\theta + 180^\circ)\right]$$

$$= 2c\left(-\cos\theta + \frac{1}{2n}\cos 2\theta\right).$$

Since the center lines of both these forces coincide we can add the forces together, which gives

$$F_4 = \frac{2c}{n}\cos 2\theta.$$

We see, therefore, that in a four-cylinder engine the primary unbalanced force is eliminated, and only the secondary unbalanced force, whose frequency is twice that of crankshaft rotation, remains. The degree of balance is therefore the same as that of the two-cylinder 90° V engine.

Vibration of Car Frame—Now let us look into the effect of this unbalanced force on the car. As the engine is generally rigidly fastened down to the frame, it is obvious that the latter will be set in vibration. Now, the forward end of the car frame can vibrate in two ways. It can rock as a rigid unit around the rear axle or the line of attachment of the rear springs to it as a fulcrum, or it may "whip" or vibrate as an elastic body.

Elastic vibration of the frame calls for a further explanation. The vibrating force in a four-cylinder engine is a reciprocating force acting in the vertical center line of the engine, or midway between the axes of the two inner cylinders. As the engine is supported in front and rear, and the force referred to acts somewhere between the supports, the engine structure itself will flex slightly under it. Generally the line of action is nearer the rear support, hence this takes the greater part of the force. The supporting member extends across the frame, and as the vibrating force acts at its middle (in most cases at least), it will be bent back and forth, whether it consists of a separate frame member or of supporting arms formed integral with the crankcase casting.

The rear engine bearer or cross member in turn rests on the frame side members, and as it alternately presses down and lifts up on these members, it will bend them back and forth.

Synchronous and Asynchronous Vibration—When bodies are set vibrating they are acted on by a force which moves them from their position of equilibrium, and by another force which tends to return them to this position. Every vibrating body has a definite period of free vibration. The period increases with the mass or weight of the vibrating body and as the restoring force decreases. Vibration may be caused either by a single impulse, in which case it will die out after a short time, or by a succession of impulses at equal intervals. It is the latter case which concerns us here. With a succession of impulses the resulting vibration differs greatly according to whether or not the period between impulses is equal to the natural period of the vibrating body. If it is equal to it we have synchronous vibration; otherwise, asynchronous.

In motor vehicles asynchronous vibration is of no consequence, as its amplitude is always very small. Synchronous vibration, on the other hand, often causes trouble. The latter form of vibration may be simply explained by means of a familiar example. A child sitting in a swing forms a pendulum that has a definite period of swing or vibration. In getting the swing going, if we time our impulses to the natural period of the swing, we obtain results with remarkably little effort, because each new impulse adds to the amplitude of swing already attained. The same principle applies in the vibration of a vehicle frame under the periodic force of the engine.

Now, the period of vibration of the front end of the car is comparatively long, of the order of 1/100 minute. In order that the vibrating force of the engine might be in synchronism with the front end of the chassis vibrating as a rigid body, since there are two impulses in each direction per crankshaft revolution (secondary unbalanced force), the engine would have to run at about 50 r.p.m. Not only is this below the operating range of four-cylinder automobile engines, but even if the engine could run at this speed, the corresponding inertia force would be exceedingly small, as this force varies as the square of the speed.

It is quite different with respect to the whip or elastic vibration of the frame around the point of intersection of the line of action of the vibrating force with its plane, as explained in detail in the foregoing. The restoring force in that case is the elastic force or spring force of the frame itself, which is comparatively great, and the weight that determines the period is the distributed weight of the frame, which is relatively small. By using a very stiff frame it would be possible to make this period so small that the speed of synchronism would be above the operating range of the engine, but with all practical four-cylinder engines the speed of synchronism is within the operating range. In one well known make of four-cylinder car now off the market it was 28 m.p.h., corresponding to an engine speed of 1500 r.p.m.

In Fig. 31 are shown the reactions on the engine frame due to the inertia forces of the reciprocating masses of a four-cylinder vertical engine. The inertia forces on the two sets of reciprocating masses are shown in dotted lines, and the resultant of the two is shown in a full line. Upward reactions on the engine frame are plotted above the horizontal axis and downward reactions below.

Lanchester Anti-Vibrator—F. W. Lanchester, an English automobile engineer, has devised a method of balancing the secondary unbalanced force, the device employed by him being known as an anti-vibrator. Two sectional views of the arrangement for a four-cylinder engine are shown in Fig. 33. To one of the short crank arms adjacent to the central main journal of the crankshaft is secured a helical gear, which meshes with another helical gear of half the number of teeth located in the bottom of the crank case transversely across the engine. This latter gear meshes with another similar one lying parallel to it. The two like gears are cut on the periphery of revolving cylinders fitted with bearing bushings and mounted on stationary studs. The cylinders are drilled each with six holes parallel to their axes, three of the holes being filled with lead and the others left empty so that the effect is the same as that of a revolving balancing

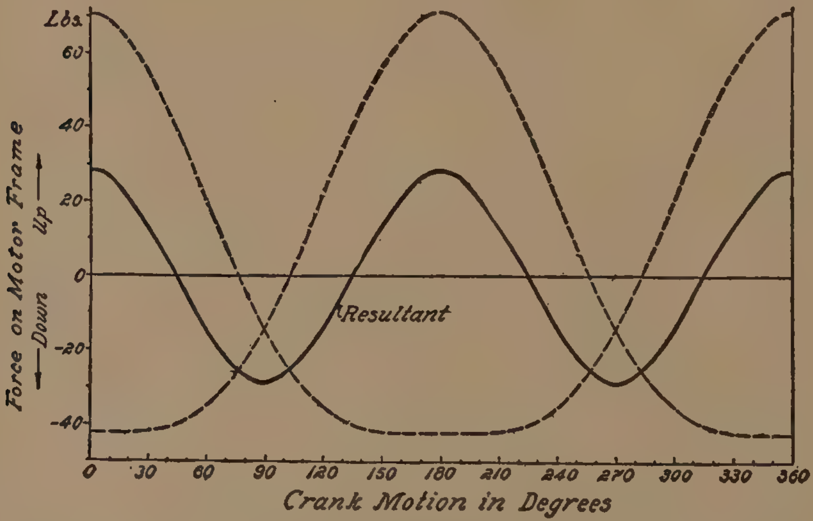


FIG. 31.—INERTIA FORCE DIAGRAM FOR FOUR-CYLINDER ENGINE.

weight. These two balancing weights revolve at equal speed (twice crankshaft speed) in opposite directions, and they are so meshed together that the horizontal components of their respective centrifugal forces neutralize each other, while the vertical components add together. Every time a pair of pistons starts on the down stroke the revolving balance weights are in the position shown in the figure, the centrifugal force on them is directed downwardly, while the secondary unbalanced force on the reciprocating parts is directed upwardly and is at a maximum. The weights are so proportioned that the two forces exactly neutralize each other.

The turning moment in a four-cylinder engine, especially at moderate speeds, is very much more nearly uniform than that of a two-cylinder engine, hence the torque reaction and the vibration due to it are very much smaller. On the whole the four-cylinder engine, when properly constructed, meets the requirement of vibrationless running fairly satisfactorily.

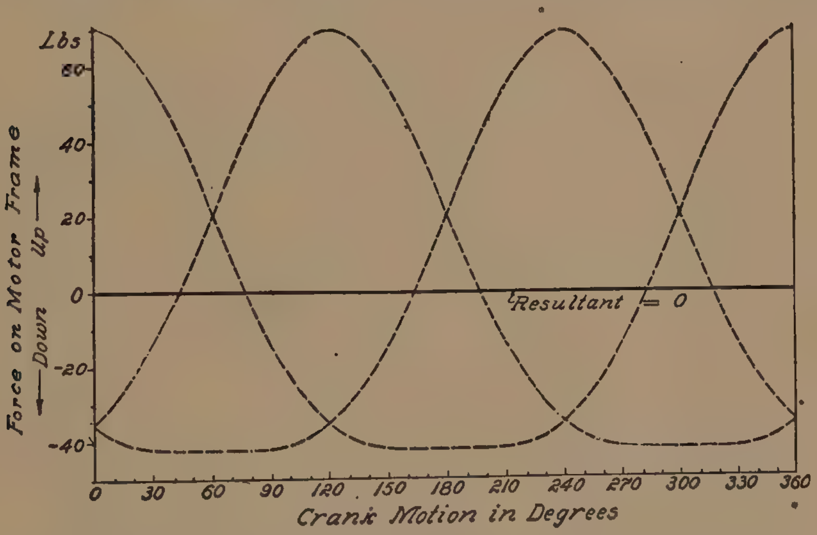


FIG. 32.—INERTIA FORCE DIAGRAM FOR SIX-CYLINDER ENGINE.

Six-Cylinder Engine—In a six-cylinder engine the pistons act on cranks set two and two together at angles of 120°, pistons Nos. 1 and 6 connecting to cranks in the same plane, also pistons Nos. 2 and 5 and pistons Nos. 3 and 4 (Fig. 30). Using the same notation as above, the inertia forces due to

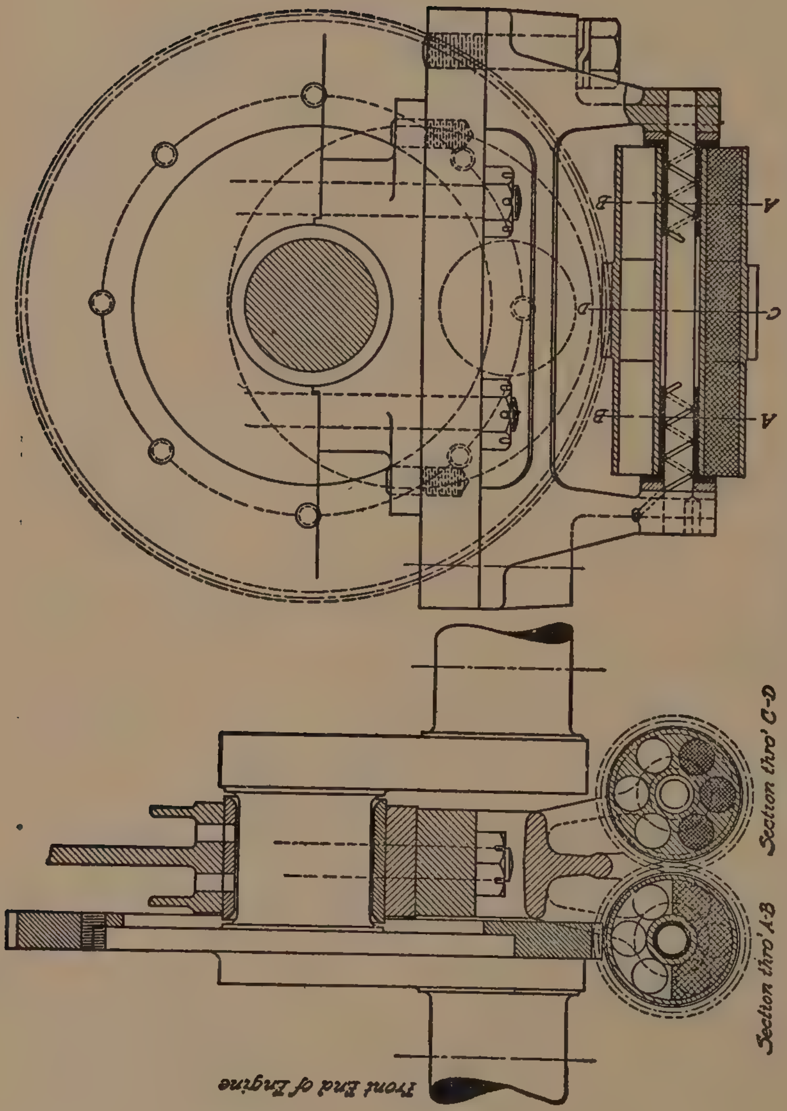


FIG. 33.—LANCHESTER ANTI-VIBRATOR FOR FOUR-CYLINDER ENGINE.

the reciprocating masses in the three pairs of cylinders, respectively, are

$$2c\left(\cos\theta + \frac{1}{2n}\cos 2\theta\right),$$

$$2c\left[\cos(\theta + 120°) + \frac{1}{2n}\cos 2(\theta + 120°)\right],$$

and

$$2c\left[\cos(\theta + 240°) + \frac{1}{2n}\cos 2(\theta + 240°)\right].$$

All act along the same center line, hence, adding them together, we have

$$F_a = 2c[\cos\theta + \cos(\theta + 120°) + \cos(\theta + 240°)] + \frac{c}{n}[\cos 2\theta + \cos 2(\theta + 120°) + \cos 2(\theta + 240°)]$$

$$= 2c[\cos\theta + \cos\theta\cos 120° - \sin\theta\sin 120° + \cos\theta\cos 240° - \sin\theta\sin 240°] + \frac{c}{n}(\cos 2\theta + \cos 2\theta\cos 240° - \sin 2\theta\sin 240° + \cos 2\theta\cos 480° - \sin 2\theta\sin 480°)$$

$$= 2c[\cos\theta(1 + \cos 120° + \cos 240°) - \sin\theta(\sin 120° + \sin 240°)] + \frac{c}{n}[\cos 2\theta(1 + \cos 240° + \cos 480°) - \sin 2\theta(\sin 240° + \sin 480°)].$$

Now, in the last equation each of the four terms in parentheses is equal to zero, hence the whole expression is equal to zero and the inertia forces are balanced. In Fig. 32 are shown curves representing the inertia forces due to each pair of reciprocating parts, and it is there proven graphically that the resultant is zero.

The variation in the crankshaft turning moment, and consequently in the torque reaction, is also much smaller than in any engine with a smaller number of cylinders, the total range of turning moment being 1.62 the average turning moment in the case of the six-cylinder, while in the four-cylinder it was 5 times the average turning moment. Consequently a six-cylinder engine may be made to run exceedingly smoothly. To insure this excellent balance it is, of course, necessary that all reciprocating parts be made of standard weight; that is, each piston must weigh the same as every other piston, and the same with respect to the connecting rods.

Torsional Vibration—When six-cylinder engines were first introduced it was found that in spite of the absolute balance of their reciprocating parts they were subject to excessive vibration at certain high speeds. The fact that vibration was limited to certain narrow ranges of engine speed made it obvious that the cause of the phenomenon lay in synchronism at these high speeds between vibration producing impulses and the natural rate of vibration of the crankshaft. It is now known that these vibrations are due to the great torsional flexibility of the long crankshaft, which makes its natural period of swing comparatively low.

By referring to Fig. 24 it will be seen that the torsional moment on the crankshaft has a regular period, equal to one-third crankshaft revolution. Obviously, the rear end of the crankshaft is restrained from rapid acceleration by the inertia of the flywheel as well as by the inertia of the parts driven by the engine. Therefore, when the crank moment increases suddenly (in the example represented by Fig. 24 from a little over 40 to about 315 lb.-ft), the crankshaft will twist slightly. As soon as the peak of the crank moment has been passed the crankshaft begins to "unwind," owing to its torsional elasticity. The next increase in the crank moment will cause the crankshaft to "wind up" again.

The torsional deflections produced by the fluctuations of the crank moment are directly proportional to the change in the moment and inversely proportional to the torsional rigidity of the crankshaft. In this connection it must be remembered that the torsional moment as represented in Fig. 24 does not act on the whole length of the crankshaft. When an explosion occurs in the rearmost cylinder, the resulting increase in torque has a twisting effect on only a short length of the crankshaft. The important point, however, is that twisting impulses are impressed on the crankshaft at regular intervals depending for their length on the engine speed. Each individual impulse will cause only a minute twisting of the shaft, but if the impulses happen to come at intervals equal to the period of free torsional vibration of the crankshaft, then their effect is cumulative and the amplitude of vibration may become very great.

Experimental Proof of Torsional Vibration—The explanation that the violent vibration of engines with long crankshafts at certain speeds was due to the torsional flexibility of the shaft was not immediately accepted by all engineers, but some experiments made at the Brooklyn Navy Yard toward the end of the War set aside all doubt on this point. The Navy had built some high-speed Diesel engines for submarines. These had eight working cylinders and two compressor cylinders in line with them, so that the crankshafts were very long. A great deal of trouble was experienced from excessive vibration at certain speeds.

In order to determine the character of the vibration, a brass disc was mounted on the free end of the crankshaft and a tuning fork, of which one prong carried a bronze reed extension with a pencil or scriber, was so arranged that if the crankshaft was turned over, the pencil would trace a spiral line on a card mounted on the disc. When the tuning fork was set in vibration and the crank revolved, the pencil traced a wavy line, with one-half the curve on each side of the spiral base line.

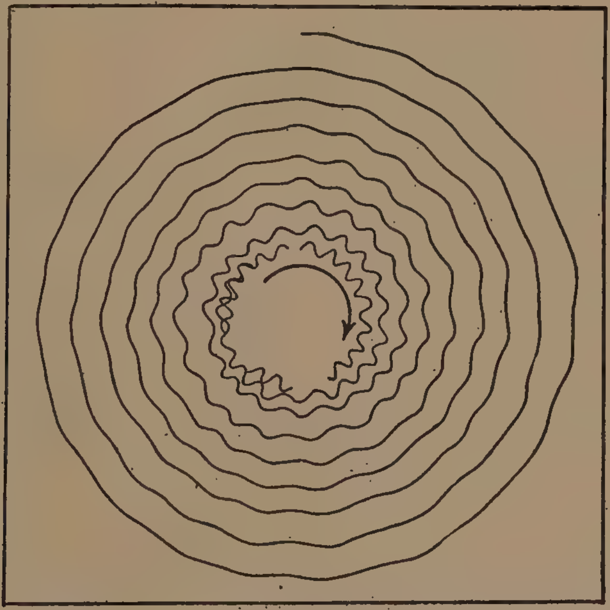


FIG. 34.—TORSIONAL VIBRATION RECORD MADE BY TUNING FORK.

Since the tuning fork has a definite rate of vibration, if the disc rotated at constant speed, the points at which the wavy line crosses the spiral base line should be spaced uniformly. As may be seen from Fig. 34, which is one of the records made with this apparatus, the crossing points alternately approach and recede, thus proving definitely that the forward end of the crankshaft alternately speeds up and slows down; in other words, that there is torsional vibration. It was found possible to trace speed and acceleration curves for the forward end of the crankshaft from this record of displacements.

Electric Torsional Vibration Indicator—An electrical instrument for recording torsional vibration was developed later by the General Motors Research Corp. and described by C. E. Summers at the S. A. E. annual meeting at Detroit in 1925. An armature having six pairs of windings is mounted rigidly on a shaft which is connected as an extension of the crankshaft. A relatively heavy brass ring, carrying permanent magnets which terminate in six poles, floats on ball bearings and is driven through weak springs. Any torsional vibration of the crankshaft results in a displacement between the windings of the armature and the poles, which movement induces an electromotive force in the windings. These windings are connected to collector rings, which, through suitable brushes, establish a circuit through an oscillograph, causing a deflection of the instrument that is proportional to the current, the current in turning being proportional to the rate of angular displacement between the poles and the windings. Suitable calibration curves enable the operator to translate the diagram into terms of frequency and amplitude of torsional vibration.

Lanchester Vibration Damper—The simplest method of eliminating trouble due to torsional vibration is to increase the diameter of crank pins and crank main journals, and the dimensions of the crank arms in proportion. This reduces the period of natural vibration of the crankshaft and brings its critical speed beyond the speed range of the engine. F. W. Lanchester has also invented a device for overcoming torsional vibration. It consists (Fig. 35) of a small fly wheel placed at the forward end of the engine, supported by the crankshaft, but not rigidly fastened thereto. It is held between friction discs, which are pressed together by springs. When the engine runs at the critical speed there will be the same tendency for the crankshaft to "thrash" as if the vibration damper were not used. But the damper has a great deal of inertia, and any sudden variation in the motion of the forward end of the crankshaft will cause the friction plates of the vibration damper to slip. Such slippage, of course, absorbs energy. Without the vibration damper there is very little resistance to torsional vibration, as steel within its elastic limit is perfectly elastic, so that any energy imparted to it setting it in vibration will keep it vibrating for a considerable time. A frictional damping device, on the other hand, will absorb the energy stored in the crankshaft by fluctuations in the torque and thereby minimize the vibration.

Eight-Cylinder Engine—An eight-cylinder V-type engine consists essentially of two four-cylinder engines placed at right angles to each other (the usual arrangement). Each of these engines has an unbalanced force in the central plane of its cylinders, as represented by Fig. 31.

We found that the expression for the unbalanced force in a four-cylinder engine is

$$\frac{2c}{n}\cos 2\theta.$$

In an eight-cylinder engine we have two such forces acting at right angles to each other. Considering an engine revolving right-handedly, looked at from the front, if we use the above expression to designate the unbalanced force due to the reciprocating parts in the left-hand set of cylinders, then the unbalanced force due to the reciprocating parts in the right-hand set of cylinders is

$$\frac{2c}{n}\cos 2(\theta + 90°)$$

$$= \frac{2c}{n}\cos(2\theta + 180°)$$

$$= -\frac{2c}{n}\cos 2\theta.$$

Now, since these two forces do not act along the same line we cannot add them directly. They act along intersecting lines, each making an angle of 45 degrees with the vertical. But the vertical components of both forces act along the same line, and so do the horizontal components, hence we may derive the horizontal and vertical components of each of these forces and then add like components together. We will call forces to the right and upward forces positive, and forces to the left and downward forces negative. We then have—

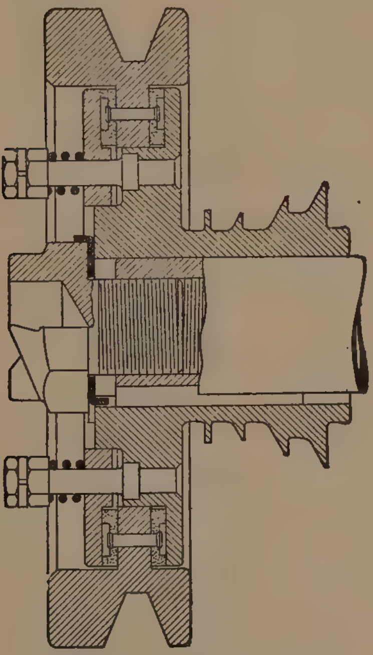


FIG. 35.—LANCHESTER VIBRATION DAMPER.

Vertical component of forces due to left-hand set of reciprocating parts

$$C_{vl} = \frac{2c}{n} \cos 2\theta \cos 45°.$$

Vertical component of forces due to right-hand set of reciprocating parts

$$C_{vr} = -\frac{2c}{n} \cos 2\theta \cos 45°.$$

The resultant of these two forces is evidently zero, hence there is no vertical unbalanced force.

The horizontal components are as follows:

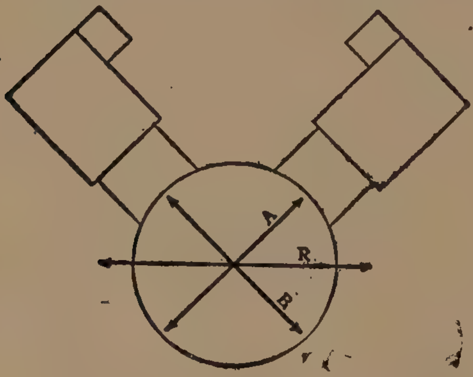


Fig. 36.—Inertia Force in Eight-Cylinder V Engine.

A—Unbalanced force of right-hand set of cylinders.
B—Unbalanced force of left-hand set of cylinders.
C—Resultant unbalanced force.

Due to left-hand set of reciprocating parts—

$$C_{hl} = \frac{2c}{n} \cos 2\theta(-\sin 45°).$$

Due to right-hand set of reciprocating parts—

$$C_{hr} = -\frac{2c}{n} \cos 2\theta\ (\sin 45°).$$

Adding the two together we find the resultant to be

$$-\frac{2c}{n} \cos 2\theta(2 \sin 45°)$$
$$= -1.414\left(\frac{2c}{n} \cos 2\theta\right).$$

Hence the resultant unbalanced force in an eight-cylinder engine is in a horizontal plane, and its maximum value is equal to 1.414 times that of a four-cylinder engine of the same cylinder dimensions (and therefore, about one-half the output). How the two inclined inertia forces combine to make a horizontal unbalanced force is shown in Fig. 36.

Using the above method of analysis, the author in 1916 (see *The Horseless Age* of February 1, 1916) made an investigation of the effect on the unbalanced forces due to changes in the angle of V. The crankshaft in all cases is supposed to be the conventional four-cylinder type, with all four throws in the same plane, the two end throws to one side of the axis and the two inner throws to the opposite side. The results are given in Fig. 37.

With a zero angle of V we have a vertical engine, of course, and the unbalanced force then is the same as in a four-cylinder vertical engine with the same reciprocating masses, being in a vertical line. When the two sets of cylinders are set to make an angle with the vertical, a horizontal component appears in the unbalanced force, while the maximum value of the force, which still occurs when it is vertical, decreases.

When the angle between cylinders becomes 60°, the unbalanced force is a constant, uniformly rotating force of the value $0.866 \times 2c/n$, using the same notation as in the previous case. The force rotates at twice crankshaft speed and therefore cannot be neutralized by the centrifugal force on balance weights rotating with the crankshaft.

Figure 37 shows that in an eight-cylinder 90° V engine the unbalanced force is a horizontal reciprocating force. Maximum values of the unbalanced forces in four-cylinder vertical, eight-cylinder 90° V and eight-cylinder 60° V engines, with the total reciprocating weights the same in all cases (each cylinder of the vertical four of about twice the displacement as each cylinder of one of the V eights) are compared in the following table:

Type of Engine	Max. Value of Unbalanced Force	Direction of Unbalanced Force
Four-cyl. vertical .	100 per cent	Vertical
Eight-cyl. 90° V...	70.7 per cent	Horizontal
Eight-cyl. 60° V..	43.3 per cent	Rotating

The advantage of the eight-cylinder 60° V engine from the standpoint of balance of reciprocating parts is offset by the disadvantage that in it the explosions do not come at equal intervals but alternately at intervals of 60° and 120°.

As the angle of V increases beyond 90° the maximum value of the unbalanced force at first increases slightly and then decreases, until when the angle becomes 180°, it vanishes. In that case we have an eight-cylinder horizontal opposed engine in which there is complete balance of reciprocating parts, provided the center lines between axes of cylinders on both sides of the crank axis coincide.

Eight-Cylinder 90° V with Two-Plane Crank—For nearly twenty years after the first eight-cylinder V engines were built the crankshafts of these engines were always of the type having all of the throws in one plane, that is, the same as used on four-cylinder vertical engines. It was shown by Archibald Sharpe in his book on The Balancing of Engines, issued in 1907, that if such an engine is built with a crankshaft with crank throws in two planes at right angles to each other, it is possible to obtain both equal spacing of explosions and complete elimination of all unbalanced forces and couples. For a long time engineers did not consider the unbalanced forces in other forms of eight-cylinder V engines of sufficient importance to warrant the use of a crankshaft with throws in two planes, which involves greater difficulty of manufacture, but in 1923 the Cadillac Motor Car Co. adopted this form of construction.

The arrangement of the crankshaft is as shown in Fig. 38. Oppositely located cylinders are exactly in the same plane and the whole engine may be regarded as made up of four two-cylinder V engines. It was shown in the foregoing that in such a two-cylinder engine the primary unbalanced forces can be entirely eliminated by the use of suitable balance weights opposite the crankpin. Thus only the secondary unbalanced forces remain to be dealt with. This is a periodic force whose period is only one half that of crankshaft rotation. That is, the unbalanced force is the same in magnitude and direction for crank positions 180° apart. Therefore, referring to Fig. 38, the unbalanced forces due to the reciprocating parts attached to cranks 1 and 4 will be equal in magnitude and direction, and their resultant will act at the middle of the length of the crankshaft. Similarly, the unbalanced forces due to the reciprocating masses attached to cranks 2 and 3 will be equal in magnitude and direction and their resultant will act at the middle of the crankshaft. However, since cranks 1 and 4 are at right angles to cranks 2 and 3, and since 90° of crank motion represents one-half period for the secondary forces, the two resultants are exactly opposite and therefore cancel each other.

The above reasoning shows that there is no rocking couple in such an engine. It is also evident that, since the resultant of the unbalanced forces of all four pairs of cylinders is at all times *nil,* there is no unbalanced free force. In our argument we assumed that there was a balancing weight on each crank throw opposite the crankpin.

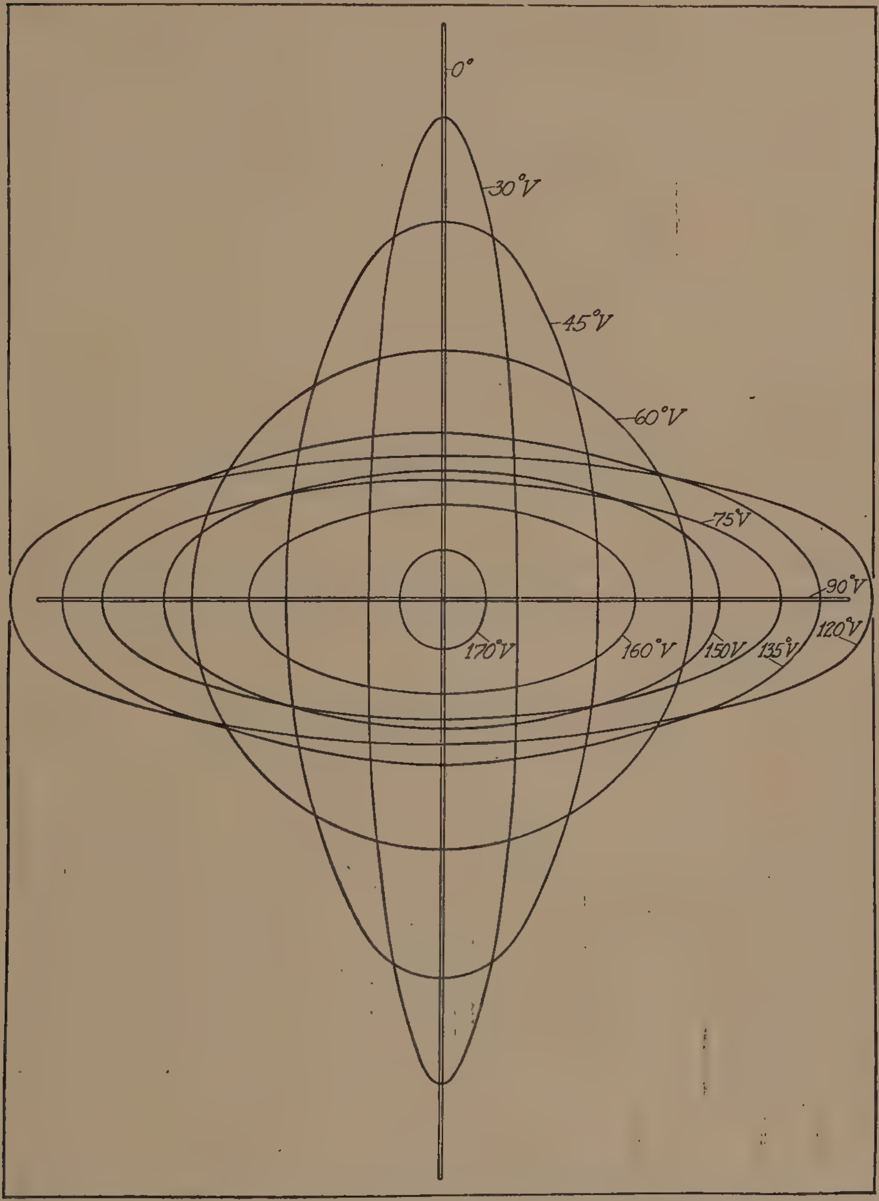


Fig. 37.—Unbalanced Forces in Eight-Cylinder V Engines with Single-Plane, Four Throw Crankshafts and Different Angles of V.

However, the balancing weights for the two throws at one end can be readily combined into one, which must be placed on the crank arm or arms between the two adjacent crank pins. This gives the construction indicated in the drawing.

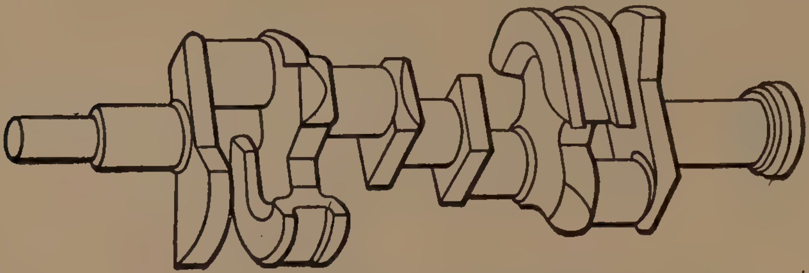


Fig. 38.—Two-Plane, Four-Throw Crankshaft for 90° V Engine.

Eight-Cylinder Vertical Engine—In eight-cylinder vertical engines an eight-throw crankshaft is used which consists essentially of two four-throw crankshafts placed at right angles to each other. Sometimes these two four-throw cranks are placed end to end, that is, all of the throws in one plane are at one end and all the throws in the plane at right angles thereto at the other end. With this arrangement, while the inertia forces in the transverse plane are neutralized, there is a rocking couple in the longitudinal

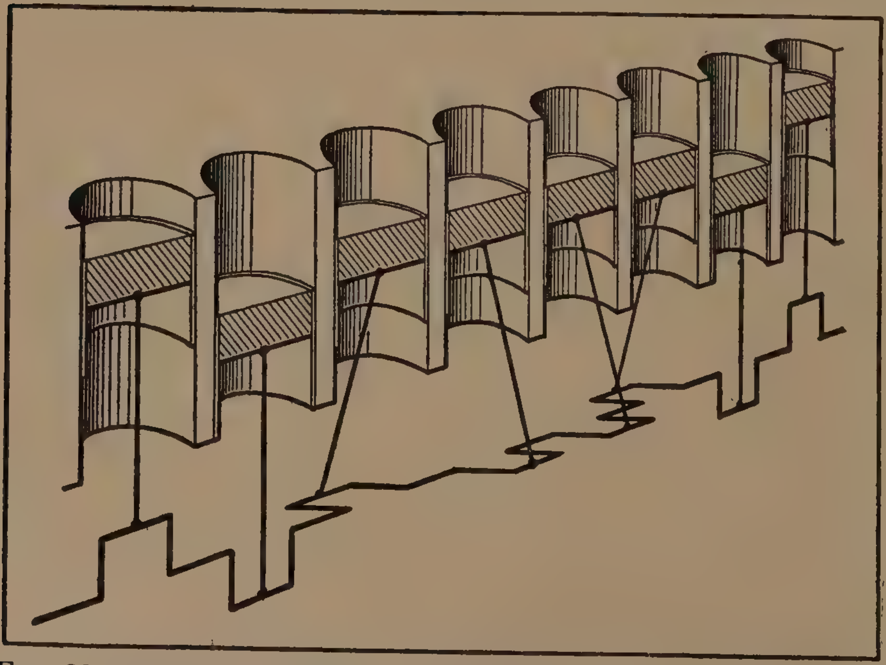


Fig. 39.—Diagram Showing Arrangement of Crankshaft for Eight-Cylinder Vertical Engine.

plane. The proper arrangement (Fig. 39) consists in using one four-throw crank for the four middle cylinders, cutting the other four-throw crank in halves and joining one half to either end of the first four-throw crank. The longitudinal centers of the two four-throw cranks then coincide, and the rocking couple is eliminated. The inertia forces are also balanced, as may be readily shown.

Let the inertia forces due to the reciprocating parts of the four middle cylinders be represented by

$$\frac{2c}{n}\cos 2\theta,$$

then those due to the reciprocating parts of the other four cylinders may be represented by

$$\frac{2c}{n}\cos 2(\theta + 90°)$$

$$= \frac{2c}{n}\cos 2\theta + 180°)$$

$$= -\frac{2c}{n}\cos 2\theta.$$

Hence the inertia forces due to the four outer cylinders are at all times equal in value but opposite in direction to those due to the four inner cylinders, and these forces cancel each other.

Twelve-Cylinder V-Engine—The twelve-cylinder V-type engine consists essentially of two six-cylinder engines set at an angle of 60°. Pistons of oppositely located cylinders act on the same crankpin, and the crankshaft is the same as in a six-cylinder engine. The angle of 60° is chosen to obtain uniformly spaced explosions. Since there are twelve explosions in two revolutions, or 720° of crank travel, for even spacing successive explosions should evidently come $720/12 = 60°$ apart. Since each six-cylinder unit of the twelve is in perfect balance, it follows that the complete engine is perfectly balanced so far as inertia forces due to reciprocating parts are concerned. The "twelve" is also subject to torsional vibration, and what was said regarding this matter in connection with six-cylinder engines applies here.

Graphical Treatment—Those who have difficulty in following the analytical treatment of engine balance as given in the foregoing will find the following graphical treatment helpful.

The lack of balance in an engine, or at least that particular kind of unbalance which we have been investigating, is due to the reciprocating parts. Let us first consider a single cylinder engine in which there is only a single set of reciprocating parts. As these start, say, from the top dead center from a standstill, a force is required to accelerate them or get them up to speed, which at high engine speeds is of the same order as the force of explosion. Now, to every action of this kind there is an equal and opposite reaction, and in this case, as the force on the reciprocating parts acts downwardly, the reaction acts upwardly on the engine structure and tends to throw it up.

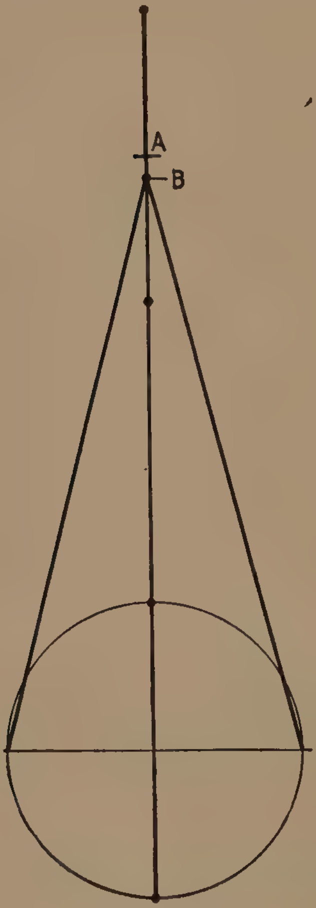


FIG. 40.

Downward acceleration of the reciprocating parts continues for about one half of the downward stroke, and the upward force on the engine structure continues for the same period, decreasing from a maximum at the beginning, to zero at the end of this period. During the remainder of the downward stroke there is negative acceleration or deceleration, and during this period there is a downward force on the engine structure, beginning from zero and attaining its maximum value at the end of the stroke. During the return stroke there is a positive upward acceleration during the first half, and consequently a downward force on the engine structure, the same as during the last half of the downward stroke. During the last half of this stroke there is negative upward acceleration, and consequently the force on the engine structure is upward, the same as during the first half of the downward stroke.

Therefore, in a single cylinder there is a reciprocating force on the engine structure, which changes direction every time the piston is traveling at maximum speed; that is, when the crank and connecting rod are at right angles to each other.

In a four-cylinder engine we have two groups of reciprocating parts, of which one is always moving up while the other is moving down (provided the cylinders are not offset). The accelerating forces, therefore, tend to balance each other, and if the motions of the two groups were so related that no change in the location of the center of gravity occurred, then there would be no force of unbalance on the engine structure. But from Fig. 40 it can be seen that there is a change in position of the center of gravity.

For the sake of simplicity we will consider only the pistons, for, although the upper portions of the connecting rod reciprocate in substantially the same manner, they simply add to the effect produced by the pistons, in the proportion of their weight to the piston weight.

When one group of pistons is at the top of the stroke the other group is at the bottom, and the center of gravity is at A (Fig. 40), midway between the centers of gravity of the upper and the lower groups. But when the crank stands at quarters, all pistons are on the same level and the center of gravity B of the whole system is then lower. It will thus be seen that during this quarter revolution of the crank the center of gravity of the pistons has traveled from A to B. To produce this motion the whole system had to have a virtual downward speed throughout the period. Owing to the inertia of the system, this speed can be acquired only gradually and must be given up gradually; that is, during the first half or so of the period (one-eighth of a crank revolution) the whole reciprocating system accelerates downwardly, and the reaction to the accelerating force is an upward force on the engine structure; during the next eighth of a revolution of the crank the reciprocating mass decelerates downwardly and there is then a downward reaction on the engine structure. We find, therefore, from Fig. 40 that as a result of the reciprocating movement in a four-cylinder engine there is a vibrating force on the engine structure parallel to the cylinder axes, and the vibrating force changes direction twice as often as in a single-cylinder engine, or is of twice the periodicity.

If we apply the same method of graphical analysis to a six-cylinder engine we find that the center of gravity of the reciprocating system remains fixed throughout the crank revolution.

The method also lends itself well to an analysis of the vibrating forces in an eight-cylinder 90° V engine. In Fig. 41 let the pistons in the set of cylinders at the right first be at the end of the stroke and those in the set of cylinders at the left at about mid-stroke. Then the center of gravity of the right-hand pistons is at A, in the highest position, and that of the left hand pistons at B′, in the lowest. The displacement of the center of gravity of one set of reciprocating masses is here somewhat exaggerated relative to its distance from the crank axis, for the sake of clearness; this does not affect the result arrived at, except that it exaggerates the scale. The center of gravity of the whole system, C, is found by connecting A and B′ by a straight line and bisecting it. Next let the left hand set of pistons be at the top and bottom of the stroke and the left-hand set at about midstroke. Proceeding as in the previous case, we find the center of gravity of the right hand reciprocating masses to be at B, that of the left-hand reciprocating masses at A′, and that of the whole system at C′, horizontally to the left of C. Therefore, in an eight-cylinder engine the center of gravity of the reciprocating masses moves in a horizontal line and there is a horizontal vibrating force on the engine structure.

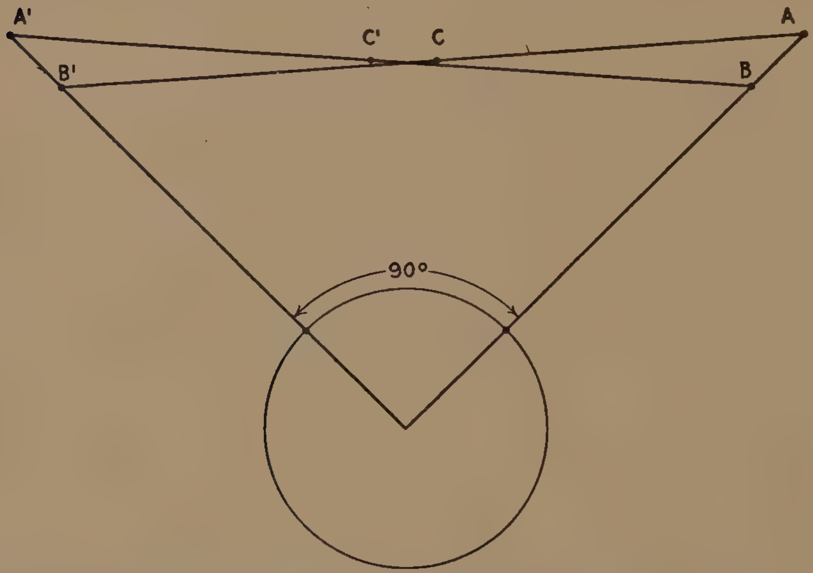


Fig. 41.—Showing Displacement of Center of Gravity of Reciprocating Parts in Eight-Cylinder 90° V Engine.

CHAPTER VI

The Cylinder

Horizontal versus Vertical Cylinders—In the early years of the automobile industry in this country, when single cylinder engines were used almost exclusively, the cylinders were generally horizontal. Later on, when the number of cylinders was increased, the vertical arrangement was adopted, because a vertical multicylinder engine requires less room on the chassis and can be arranged more accessibly and more symmetrically. It also possesses certain advantages with respect to lubrication. It was often urged against the horizontal engine that the weight of the piston had a tendency to wear the cylinder bore oval, but it can easily be shown that in comparison with the side thrust of the piston on the cylinder wall due to the angularity of the connecting rod, the piston weight is absolutely negligible, and this feature had nothing to do with the abandonment of the horizontal type.

Arrangement of Valves—The cylinder of a gasoline engine may be given many different forms, the differences depending chiefly upon the location of the valves. In designing a cylinder or group of cylinders, there are two main factors to be kept in view, viz., the form of the combustion chamber and the means of actuating the valves from the crankshaft. One object to aim at is to get a combustion chamber with the least wall surface in proportion to its cubical content, as the loss of heat through the walls depends very largely upon the extent of this surface. This consideration would lead to the adoption of a spherical combustion chamber, for of all geometrical forms the sphere has the least surface area for a given volume. This ideal form is approached in certain Knight sleeve valve engines. With poppet valves the nearest approach to an ideal combustion chamber is obtained when both inlet and exhaust valves are located in the cylinder head at an angle of 30° to 45°, the cylinder head being made of hemispherical form, or domed. The piston head can, of course, be dished as it is in the Silent Knight engine, but this is seldom done.

The next best arrangement from this point of view is that denoted by 3 in Fig. 42, where both valves are located in the cylinder head side by side and parallel. This gives a cylindrical form of combustion chamber. Next follow the arrangements 2, 4 and 6, which are about equal. In arrangement 2 the inlet valve is located directly above the exhaust valve in a side pocket and opens downwardly. In arrangement 4 the inlet valve is located in the center of the cylinder head, and the exhaust valve in a side pocket, while in arrangement 6 the two valves are located opposite and in line with each other in a valve pocket formed on the cylinder head. In each of these cases there is a valve pocket the size of which is determined by the size of one valve head. These pockets naturally add considerably to the wall surface for a given combustion space volume. Next in order of combustion space wall area comes the arrangement 7, with both valves in a single pocket on the side of the cylinder, and finally that denoted by 1, with the inlet and exhaust valves in side pockets on opposite sides.

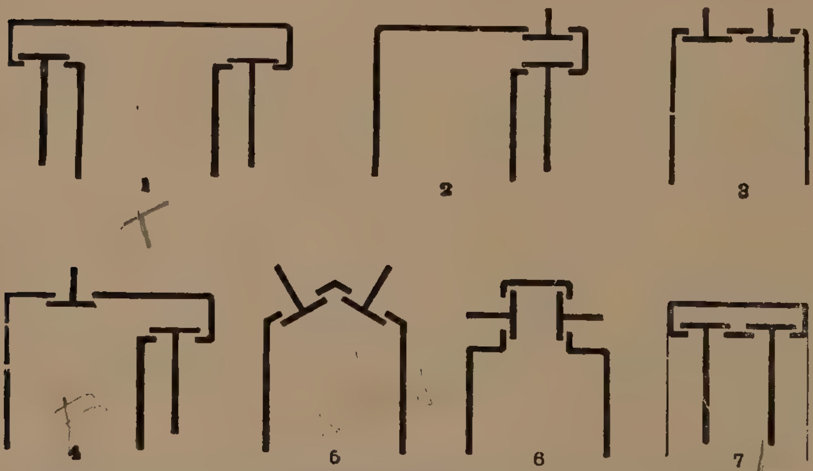


Fig. 42.—Valve Arrangements and Cylinder Types.

Minimum wall area is, however, only one consideration in the design of a cylinder, and, in the estimation of many designers, by no means the most important. Upon the arrangement of the mechanism for actuating the valves depends to a large extent the more or less noiseless operation of the engine, as well as its general appearance, and, to some extent, its ability to run at high speeds and to give a large output in proportion to its dimensions. This mechanism is of the simplest form when both valves are located in a single side pocket, in which case only a single camshaft for operating the valves, and a single set of camshaft gears are needed. The camshaft is placed directly underneath the valve chamber and the valves are lifted by direct thrust. With the valves in pockets on opposite sides the method of operation is similar, but two camshafts and two sets of camshaft gears are required. When the valves are arranged in an inverted position, as when they are located in the cylinder head or when the inlet valve is located above the exhaust valve, a more complicated intermediate mechanism is required, with more chances for developing play at the joints. Sometimes in multicylinder engines the camshaft extends centrally across the top of the cylinders, being driven from the crankshaft through an intermediate vertical shaft or through a silent chain, but more frequently there are a series of vertical valve rods at the sides of the cylinder, which are moved upward by the cams, the direction of motion being reversed by short tappet levers whose arms bear against the valve stems and the valve rods respectively. From the viewpoint of simplicity of valve gear, arrangement 7 (the L-head cylinder) is undoubtedly the most advantageous. Arrangement 1 (the T-head cylinder) permits of using larger valves, but owing to its need of two camshafts for actuating the valves and its greater proneness to detonation it has been almost abandoned for automobile work.

Valve Location Practice—It is generally agreed that the valve-in-head cylinder is capable of developing more power per unit of displacement than any other type, and this has led to its practically universal use for aircraft and racing engines. Giovanni Enrico, a prominent Italian engineer, at one time placed this advantage at 20 per cent. Another point in favor of this type of engine is that the entire combustion chamber can be readily machined, which permits of accurately equalizing the compression volumes in the different cylinders, and of polishing the walls of the combustion chamber with a view to preventing the accumulation of carbon and consequent pre-ignition.

Other considerations that come up in the determination of the most suitable valve arrangement are the attachment of the admission and exhaust pipes and the easy enclosure of the valve mechanism. In weighing the advantages and disadvantages of the various arrangements against each other, different designers naturally reach different conclusions. Of the passenger-car engines in production by American makers in 1925, 73 per cent had L-head engines, 21 per vent valve-in-head engines, 3 per cent T-head engines and 3 per cent other types (one valve in the head and the other in a side pocket, etc.). Ten years earlier, in 1915, the percentage of L-head engines was practically the same, while the positions of the valve-in-head and T-head types were practically reversed; at that time 20 per cent were T-head engines and only 5 per cent valve-in-head engines. European designers turned to the L-head type of cylinder—sometimes referred to as the Renault type—earlier than we did, and at the first post-War shows, held in Paris and London in 1919, the T-head engine was entirely absent.

Another factor that affects the problem of valve location and form of cylinder head is the effect on the tendency of the engine to detonate. Proneness to detonation is generally believed to be a function of the distance from the point of ignition (the spark plug points) to the most remote part of the combustion chamber. Hence, from this point of view the valve-in-head cylinder with its compact compression chamber, has the advantage over the L-head cylinder, while the latter in turn is superior to the T-head cylinder. Many engineers are of the opinion that for operation on kerosene, which fuel often gives trouble from detonation, the valve-in-head cylinder is better than any other.

Form of Combustion Chamber—Some engineers attach great importance to so designing the combustion chamber that the combustible charge in it is in a state of turbulence at the moment it is ignited. It was discovered by Dugald Clerk that the rate of flame propagation through a combustible charge ignited in a glass tube can be materially increased by agitating the mixture by means of a fan inside the tube. Quick combustion of the charge early in the expansion stroke is desirable, because the earlier in the stroke the heat of the fuel is liberated, the more chance it has to be converted into mechanical work.

It has also been found that with a form of combustion chamber engendering considerable turbulence, there is less tendency for the engine to knock or detonate, and in consequence a higher compression ratio can be used, which in turn results in a higher thermal efficiency. These two results of turbulence, viz., increase in the rate of flame propagation and suppression of detonation, are not necessarily related. The latter may be explained as follows:

Effects of Turbulence—While the charge is being compressed in the cylinder its temperature rises, the heat causing this temperature rise being derived partly from the work of compression and partly from contact with the hot cylinder walls. If the temperature of any part of the charge becomes sufficiently high previous to the occurrence of the ignition spark, self-ignition takes place, and self-ignition seems to be one of the conditions of detonation or knocking.

Now, the temperatures of different parts of the combustion chamber wall will be different, and if the charge in the combustion chamber were in a state of relative immobility during the compression stroke, that portion in contact with the hottest part of the wall naturally would have a better chance of reaching the self-ignition point than any part of the charge if the latter were subject to turbulence.

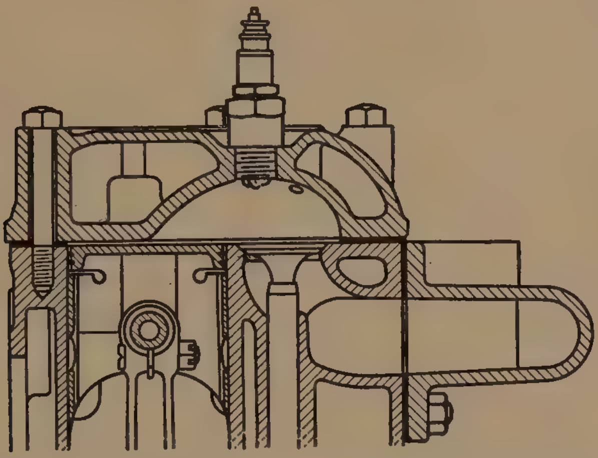


Fig. 43.—Turbulence Type of Cylinder Head.

Turbulence in the combustion chamber and minimum wall area are conflicting requirements. Various shapes of combustion chamber have been proposed with a view to producing turbulence, but only the one illustrated in Fig. 43 has been extensively used. A patent on this form of combustion head was issued to Harry Ricardo, a British automobile engineer.

Cylinder Grouping—In passenger-car engines all cylinders with their axes in the same plane are now always made in a single casting (or cast *en bloc*, as the French term it). Trucks and tractors are generally equipped with four-cylinder vertical engines, and most of these engines also have all their cylinders cast in one block, though some of the larger have pair-cast cylinders. When engines with four cylinders were first introduced the cylinders were usually cast either singly or in pairs, owing to the technical difficulties involved in producing the complicated multiple cylinder castings. Another reason was that the expense for repairs would be less, in case a cylinder jacket was damaged by frost or a cylinder bore injured by scoring.

Block cylinders took the place of those individually cast because with the former the overall length of the engine is less, the engine structure can be made more rigid for the same weight, the tubular connections to the engine are greatly simplified and the amount of machine work required on the engine is reduced.

As regards simplification of the connections, in a four- or six-cylinder block engine, the jacket spaces of all cylinders are in one, hence there need be only one inlet and one outlet connection for the cooling water. It is even possible to cast the inlet and exhaust manifolds in the cylinder head or the cylinder block, in which case only a single inlet and a single exhaust connection are needed, which makes for a very clean-cut design. This plan is not very popular with American designers, but a good many Continental cars have the inlet manifold entirely within the cylinder casting, the carburetor being bolted directly against the cylinder block. A further advantage of block construction resides in the fact that a housing for the valve springs may be cast integral with the cylinder block, so that by merely fitting a cover plate the entire valve mechanism may be neatly encased.

Integral Cylinder Heads—In Europe the cylinder block with integral heads is still used to a certain extent, although it is passing there too, the same as it has passed in this country. Casting the head integral with the cylinders obviates the necessity of finishing the joint faces and does away with the gasket, but it makes the cylinder block a much more difficult casting and has other disadvantages.

In the case of a block of L-head cylinders each valve is introduced through an opening in the cylinder head slightly larger in diameter than the valve head, which is closed by means of a threaded plug generally referred to as a valve cap. This plug presents to the hot gases within the combustion chamber a considerable surface which is not water-cooled, and is therefore likely to form a hot spot. It is usual to screw the spark plugs into the caps over the inlet valves.

Valve Cages—With valve-in-head engines, if the cylinder head is cast integral the valves are generally located in cages which are inserted into suitable openings in the cylinder head. The cage contains the valve seat, and as the heat absorbed from the hot gases by the valve head must pass, for the most part, first through the valve seat into the cage and then through the seat of the cage into the metal of the head, the cooling of the valves is not nearly as effective as where the valve seats are in the metal of the cylinder head and are directly water-cooled. This is of particular importance in the case of the exhaust valve. The valves therefore become hotter, and in order to avoid trouble from detonation it is necessary to work with a lower compression pressure than where the valve seats are water-cooled directly.

Separate Cylinder Heads—When the cylinder heads are a separate casting the cylinder and jacket cores can be more securely supported in the mold, and the cylinder castings will be more nearly true to pattern, with the result that after the cylinder is finished the cylinder walls will be more nearly uniform in thickness.

With an engine having a removable head it is possible to thoroughly clean the combustion chamber of carbon, by scraping, after the head has been removed. If it is desired to locate the valves in the head, which, as explained before, is a very desirable arrangement from some points of view, they may be seated directly on a water-cooled surface if the head is removable. It is easier to machine the combustion chamber wall over its entire surface.

With a separately cast cylinder head it is, of course, necessary to provide a joint between the cylinder and the head which is gas-tight under a pressure of several hundred pounds per square inch, and provision must also be made for placing the water jacket around the cylinders proper in communication with the jacket of the head casting. It was the difficulty of making and maintaining a gas-tight and water-tight joint that led manufacturers for a good many years to give preference to engines with integral cylinder heads.

In the early engines a gasket cut out of asbestos cloth was used. Segmental openings were cut in the gasket through which the cylinder jacket space communicated with the head jacket space. The gaskets were smeared with an oil and graphite mixture on one side and a solution of shellac in alcohol on the other, so that if the joint was broken all of the gasket would adhere to the same side and the gasket remain intact. But as the wall thicknesses were made smaller and smaller it became constantly more difficult to keep the gaskets in place by the pressure between cylinder and head. The clamping surfaces were faced off with a coarse feed and the gasket material was rendered more tenacious by interweaving it with fine copper wire; nevertheless the gaskets would frequently blow out under the pressure of explosion or burn out when the water circulation failed, and renewing them was one of the most dreaded repair jobs in those days. The result was that about 1910 nearly every automobile engine had its cylinder heads cast integral with the cylinders.

Copper-Asbestos Gaskets—It was then that the copper-asbestos gasket arrived and caused a gradual reversion in practice. This consists of an asbestos sheet cut or stamped to the desired form, which is armored with thin sheet copper. There is a copper sheet on each side of the asbestos sheet and the two copper sheets lap along the edges of the asbestos, so that the latter is completely encased. In these copper-asbestos gaskets the copper provides the tenacity, while the asbestos provides the compressibility needed in a packing material.

In order to make a tight, permanent joint with a copper-asbestos cylinder gasket it is necessary to make provisions for an adequate number of studs, distributed as nearly uniformly as possible. With L-head cylinders from 18 to 20 studs are used for a four-cylinder block, from 24 to 26 for a six-cylinder and from 30 to 32 for an eight-cylinder. With valve-in-head cylinders only two rows of studs are required, instead of three, and the total number is therefore less, viz., 12 for a four-cylinder block, 16 for a six-cylinder and 20 for an eight-cylinder. To prevent distortion of the casting by drawing up the nuts, there must be plenty of metal in the bosses for the studs, and the studs must not be too near the valve seats. In the design of the heads, careful attention must be given to the avoidance of pockets which might form steam traps. It is not necessary to use very large water ports. Moderate sized ports judiciously distributed are better, as they make it easier to prevent leaks. A typical four-cylinder gasket is shown in Fig. 44.

Plugs over the valves are not used with removable heads. With this type of engine the operations of grinding in of valves and removal of the carbon deposits in the combustion chamber walls are usually combined, and this warrants the removal of the heads.

Where the cylinder head is detachable the compression chamber may be formed either in the cylinder block or in the head. In L-head cylinders it is practically always formed in the head, while in valve-in-head cylinders it is formed in the cylinder block in the majority of cases. From the piston displacement given by the bore and stroke and the compression ratio decided upon, the necessary compression space can be readily calculated, and in laying out the design care must be taken to obtain this compression volume accurately. With an irregular compression space this is sometimes a rather difficult problem, and it is then a good plan to have a patternmaker make an accurate model in wood of the compression chamber. By weighing this model and also a one-inch cube of the same wood, the volume of the compression chamber can be calculated.

Cylinder Material—Automobile engine cylinders are generally cast from close-grained gray iron approximating the following composition:

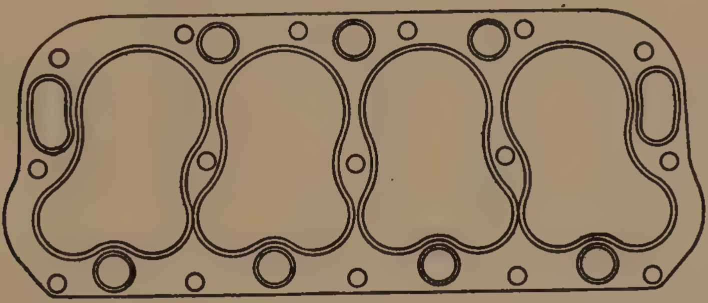


FIG. 44.—CYLINDER HEAD GASKET FOR FOUR-CYLINDER ENGINE.

	Per Cent
Carbon	3.25
Silicon	2.00
Phosphorus	0.75
Manganese	0.50
Sulphur—not to exceed	0.10

Such iron has a minimum tensile strength of 24,000 lbs. per sq. in. and pours well. The use of steel for cylinders has often been suggested, and for racing and aircraft engines cylinders are sometimes made from hollow steel forgings and even from solid billets. Several American manufacturers have used cylinder castings of semi-steel, more properly called high-tensile cast iron. This material is made by adding a certain percentage of scrap steel to the melt of cast iron, which results in a finer grain and in somewhat better tensile properties. At one time vanadium iron was used to some extent. Vanadium present in small quantity in the iron ore accomplishes the same result as when incorporated into steel; that is, it eliminates some of the objectionable impurities, acting as a "purger."

Following are the compositions of domestic and foreign cylinder irons respectively, as required by the specifications of one large American concern:

DOMESTIC IRON CASTINGS

	Per Cent
Silicon	1.25 to 1.60
Sulphur .....less than	0.11
Phosphorus	0.50 to 0.80
Manganese	0.50 to 0.80
Combined carbon .....over	0.40
Graphitic carbon .....under	3.00

IMPORTED CYLINDER CASTINGS

	Per Cent
Silicon	2.50 to 3.00
Sulphur .....under	0.10
Phosphorus	1.50 to 1.80
Manganese	0.30 to 0.50
Combined carbon .....over	0.50
Graphitic carbon .....under	3.00

Specifications for Cylinder Castings—Among the other points covered by these specifications are the following: "Castings must be made from hard, close-grained iron; free from shrinkage cracks, spongy spots, blow-holes and foreign substances; true to pattern and must not develop any defects in machining. Drillings taken from several castings from each day's cast for analysis must show the above composition. When specified on the drawing, the maker's brand or trademark and date of pouring must be cast in raised figures on every casting at the place designated on the drawing. The order and requisition numbers must be painted on each casting with white lead. The manufacturer must furnish two test pieces 1¼ inches in diameter by 14 inches long to represent each day's work. The date upon which these are cast must be shown in raised figures on each test piece. The test pieces should be cast at different times during the pouring period, and the analysis of the test pieces must agree (within reasonable limits) with castings made that day. Castings not within the weight limits given on the drawings and failing to meet the above requirements will be rejected and returned."

Nickel-chromium Irons—To make it possible to successfully cast a multiple cylinder block with thin walls, the iron must pour well and have a "long life" (as the foundrymen call it). These characteristics are strengthened by a high phosphorus content, but, unfortunately, this element tends to make the iron soft and less resistant to wear.

Certain iron ore mined in Cuba contains small percentages of nickel and chromium, and the metal made from this ore, known as Mayari iron, is sometimes added to gray iron for cylinder castings. Mayari iron therefore is a natural alloy. It is claimed that it is free from oxidation and has a lower solidification point, and that the "longer life" of the iron improves the "feeding" of castings when they are properly gated, in spite of the low phosphorus content. Castings when sectioned show sound metal even where there are heavy bosses and thick sections. Cylinder castings made of a mixture containing 10 per cent of Mayari iron showed a tensile strength of 36,740 lbs. per sq. in., according to the makers of the iron; a transverse strength of 4250 lbs. and a Brinell hardness of 223–229. The same iron is also used for cylinder heads and pistons. Mayari iron is produced by the Bethlehem Steel Co.

Quite recently similar results to those obtained from Mayari iron have been sought by the addition of a small quantity of nichrome to the iron in the ladle. Nichrome is an alloy made by the Rider-Harris Co. and has the following approximate composition: 60 per cent nickel, 12 per cent chromium, 24 per cent iron, and small percentages of carbon and other elements. Metallic nichrome is put into the hot but empty ladle and the iron is then added. It is claimed that this addition renders the iron considerably harder and more nearly uniform in texture. The effect on hardness of various proportions of nichrome is shown by the following table:

Percentages of Nichrome ..	1	2	3
Brinell hardness	207–217	223–235	241–255

Following is the analysis of an iron to which nichrome had been added:

	Per Cent
Total carbon	3.14
Silicon	2.07
Manganese	0.55
Chromium	0.42
Nickel	1.57
Copper	0.10

Among those who have used alloyed cast iron are the Cadillac Motor Car Co. and International Motor Co., manufacturers of Mack trucks.

Cylinder Stresses and Wall Dimensions—In calculating the maximum stresses which may occur in a gasoline engine cylinder, account must be taken of the fact that if one or more charges should be "missed" (i.e., fail to ignite) all of the spent gases will be pumped out of the cylinder, and the next charge to fire will be stronger in the proportion of the piston displacement plus compression volume to the piston displacement. Therefore, although the normal explosion pressure in an automobile engine averages about 300 lbs. per sq. in., it will be well to figure on abnormal explosion pressures of 400 lbs. per sq. in.

Now, consider a section of a cylinder of b inches bore 1 inch long (Fig. 45). The pressure developed in the cylinder by the explosion tends to rupture the cylinder wall along lines parallel with the center line and at opposite ends of a diameter. The rupturing force for the section of the cylinder considered is evidently $400b$ lbs. (considering the explosion pressure to be 400 lbs. per sq. in.). If the cylinder wall has a thickness t and its material has a tensile strength of 24,000 lbs. per sq. in., the resistance to rupture of the two sections 1 inch long and t inch thick is

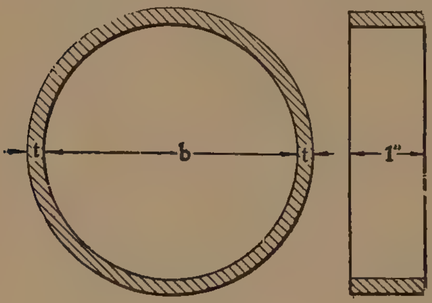


Fig. 45.

$$2 \times t \times 24{,}000 = 48{,}000t \text{ lbs.}$$

Calling the factor of safety f, we have

$$f = \frac{48{,}000t}{400b} = 120\frac{t}{b}.$$

If we allow a factor of safety of 4, then

$$4 = 120\frac{t}{b},$$

and

$$t = \frac{b}{30} \text{ inch.}$$

This formula when applied to cylinders of small bore gives values for the cylinder wall thickness which, while large enough so far as withstanding the stresses of a normal explosion is concerned, would be too small from the standpoint of shop production. If the water jacket is cast integral, as it usually is, the cylinder can be machined only on the inside, and the minimum thickness of the wall then depends upon the accuracy with which the cores are set. Some allowance must be made for inaccurate core work. From the data of a large number of modern engines of both American and European construction the following formula for wall thickness has been derived:

$$t = \frac{b}{30} + \tfrac{1}{8}\text{ in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (38)$$

This formula can be safely applied to the whole range of sizes of automotive engines with cast iron cylinders. It should be pointed out, however, that the assumption of a maximum explosion pressure of 400 lbs. per sq. in. is justified only in the case of normal combustion. Tests made at the Bureau of Standards at Washington have shown that when an engine knocks or detonates badly, the pressure may exceed 1300 lbs. per sq. in. But the addition of ⅛ in. to the calculated thickness and the fact that when the pressure is at a maximum only a short length of the cylinder is subjected to it, ensures an adequate factor of safety even under these abnormal conditions, under which the engine should never be run, at least not for any length of time.

In using equation (38) to determine the necessary cylinder wall thickness, the nearest size in thirty-seconds of an inch to the result given by the formula should be chosen. If the cylinder becomes worn in the course of use, or damaged by lack of lubrication or on account of the piston pin coming adrift, it may be saved by regrinding. In the case of ordinary wear an increase in the diameter of 0.010 inch is usually sufficient, but in cases of scoring the diameter must sometimes be enlarged as much as 0.040 inch. Some manufacturers make it a practice to furnish over-sized pistons to be fitted into reground cylinders, and the Society of Automotive Engineers has standardized these pistons, which are made 0.010, 0.020, 0.030 and 0.040 inch over-size. Wall thicknesses as given by the above formula admit of such regrinding.

The cylinder head is usually made of the same thickness as the cylinder wall. If it were of hemispherical form its factor of safety would be the same as for the cylinder wall. Wherever possible the head is rounded, but where it must be made flat it is well to make the thickness somewhat greater than that of the cylinder wall as determined by equation (38), or else to stiffen the head by ribs or by bosses tying it to the jacket wall.

Jacket Dimensions—The water jacket usually extends down the cylinder to a point even with the top edge of the piston when in its lowest position. Some make the jacket shorter, while a few make it longer. Where it is carried down farther than the bottom of the stroke, the object usually is to avoid the offset at its lower end, by making the crankcase wall a continuation of the jacket instead of the cylinder wall. This gives a simpler outside form.

It is obvious that a great deal more heat passes into the jacket above the compression chamber and at the upper end of the cylinder wall than at the lower end of that wall, because the first-mentioned portions of the wall are subjected to the heat of the burning gases throughout the power stroke, and they are exposed to the maximum temperature occurring within the cylinder, which is not the case with the lower end of the cylinder wall. Some designers therefore taper the jacket down from the top to the lower end, so as to place a larger body of water around the compression chamber, where most of the heat must be absorbed. In most engines, however, the depth of the water jacket is uniform from top to bottom.

This depth varies somewhat in different designs, and no relation between the width of the water space and the bore of the cylinder is recognizable in the data at hand. The widths vary between 7/16 and 3/4 inch. A good rule would be to make the jacket 1/2 inch wide for cylinders from 3 to 4 inches, 5/8 inch for cylinders above 4 and up to 5 inches and 3/4 inch above 5 inches. Liberal water spaces have the advantage that the core sand can be more effectively removed from the casting.

The jacket wall is generally made as thin as the foundry process permits of. It can, of course, be made thinner in a small cylinder than in a big one, because in the former the area is smaller. Average practice with regard to cylinder wall thickness is as follows:

Cylinder bore, inches	3	4	5	6
Thickness of jacket wall	5/32	3/16	7/32	1/4

On top of the cylinder a liberal water space is generally allowed. The under side of the cylinder head in an L-head engine usually is rather irregular, while the top is made plane or slightly rounded, and the differences in total thickness are taken up in the water jacket, the depth of which varies from point to point. In a valve-in-head engine the water jacket must be carried all around the exhaust passages, and in that case the jacket, therefore, is of even greater depth and more complicated form.

With valve-in-head cylinders the location of the water outlet presents some difficulty, because the valve mechanism on top of the engine is usually provided with a cover. One solution of the problem consists in forming a number of outlet bosses on the head over to one side, so they come outside the valve cover, using a water return manifold. While this tends to promote uniform circulation, it makes for dissymmetry, which is the more objectionable because the manifold is located very prominently on top of the engine. The more common plan is to have an outlet at the front end of the head, just outside the valve cover and usually oblong in form, with the long diameter across the engine, so as to minimize the overhang.

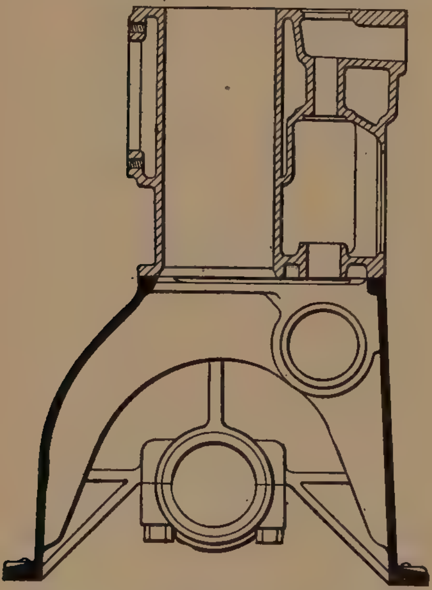
In those engines in which the jacket extends only to the lower end of the piston stroke the cylinder wall is continued of the same thickness below the jacket.

Relation to Crankcase—The cylinder block may be bolted down so as to be entirely above the crankcase or it may be sunk into the latter more or less. The former arrangement, illustrated in Fig. 46, may be regarded as conventional or standard where the block and crankcase are cast separately. Accurate registry of the cylinder bores with the openings in the top surface of the crankcase is assured by the use of locating holes in both of the facing surfaces, from which practically all machining operations on both parts are located and which are used as dowel holes when assembling the parts.

In order to obtain a simpler and smoother outside form, that part of the cylinder below the water jacket is sometimes sunk into the crankcase, or, looking at the subject from a different angle, the crankcase is carried up to the water jacket, as illustrated in Fig. 47. In some designs, including the German Maybach, the British Triumph and the American Waukesha bus engine, almost the whole of the cylinder block is sunk into the crankcase, which latter is carried up to the valve chamber of the L-head cylinders. The object in this case evidently is to make as much of the engine structure as possible of the lighter aluminum alloy and as little as possible of cast iron. The Maybach design is shown in Fig. 48.

One disadvantage of this latter construction is the difficulty of finishing the bearing surface on the cylinder block where it fits against the crankcase casting, which evidently can be done only by running a milling tool all around the block, instead of straight across it, as with the conventional block. This would not commend itself to American production engineers.

In the Triumph the cylinder block is secured to the crankcase by means of studs screwed into the valve chamber from below, the nuts for which come inside the valve spring chambers in the aluminum casting, while on the opposite side of the block collared studs are used which are passed through from the top and serve to hold both the crankcase and the cylinder head. In the Maybach, long studs pass entirely through the cylinder block and the crankcase and hold in place both the cylinder head and the crankshaft main bearing caps, two of the bolts in addition holding the oil-pump bracket below one of the main bearings.



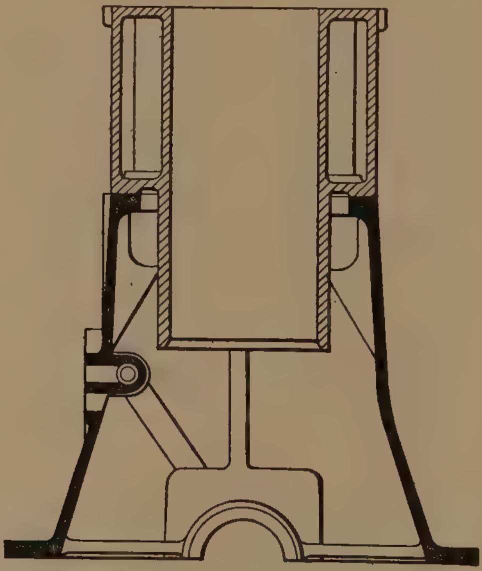


Fig. 46 (left).—Cylinder Block Bolted to Top of Crankcase.
Fig. 47 (right).—Cylinder Block Extending into Crankcase.

Joint with Crankcase—Those cylinder blocks which set right on top of the crankcase are provided with a flange which is of about one and one-half times the thickness of the cylinder wall and of the same or slightly greater width. On the valve side in an L-head cylinder this flange is extended to form the bottom or floor of the valve chamber. Lugs of a depth about twice the thickness of the flange are provided for the cylinder block holding-down bolts. These are usually placed at each of the four corners of the block and also at each side in the center plane between adjacent cylinders. It is in these planes that the partition walls of the crankcase are located, and by providing suitable bosses in these walls, a very secure anchorage can be obtained.

In a few cases so-called "through" bolts are used, which, in addition to holding the cylinder block down to the crankcase, also hold the caps of the crankshaft main bearings in place. An example of this practice, which was taken over from the aircraft engine field, is shown in Fig. 49. For the cylinder blocks of passenger-car engines, of 3–3½ in. bore, studs or cap screws of ⅜–7/16 in. diameter are used, and proportionately larger or smaller ones for other bores.

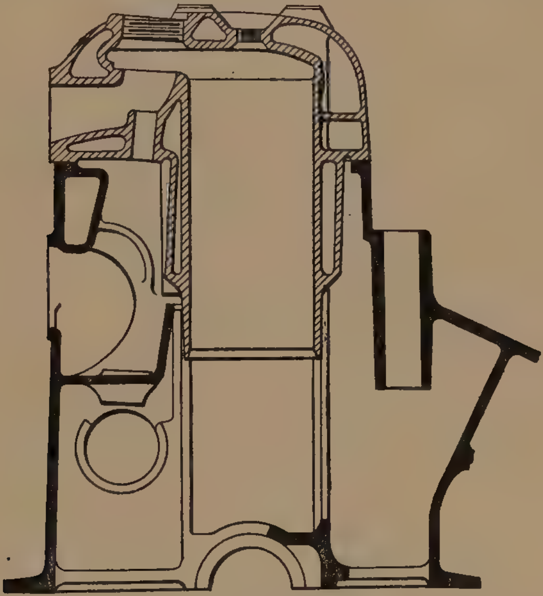


FIG. 33.—LANCHESTER ANTI-VIBRATOR FOR FOUR-CYLINDER ENGINE. CRANKCASE.

Common Cylinder Wall—In the design of multiple cylinder blocks, about the first question that comes up is whether it is necessary to have a water space between the cylinders at the point where they are nearest to each other or whether a common wall may be used at this point. At one time, when the cylinder bore determined the overall length of the engine, the common wall was in extensive use, especially in Europe, but today by far the greater number of cylinder blocks have a water space between the cylinders. With a common cylinder wall it is necessary to provide an extra thickness of metal where the wall is common to both cylinders. By referring to Fig. 50 it will be readily seen that the thinner this wall is made the greater will be the distance from the center of it to the water jacket; and, besides, the cross section of the metal through which the heat must travel will be less, while the area through which the heat is absorbed will be greater. A good rule is to make the minimum thickness of the common wall twice the regular cylinder wall thickness. The best plan, however, is to carry the water jacket completely around each cylinder.

Valve Seats—Water jackets should preferably be carried close to and all around the valve seats. It has been shown that the heat absorbed by the valve heads through their contact with the burning gases passes off chiefly through the seats, and if the water comes close to the seats the cooling of the valve heads is more effective, while if it extends all around the seats the cooling of the heads is likely to be more nearly uniform, which prevents warping of the heads. In valve-in-head engines, however, if the valves are arranged with their axes in the fore-and-aft center plane, it is generally not possible to provide a water jacket between the inlet and exhaust valve pockets, as this would add unduly to the length of the engine.

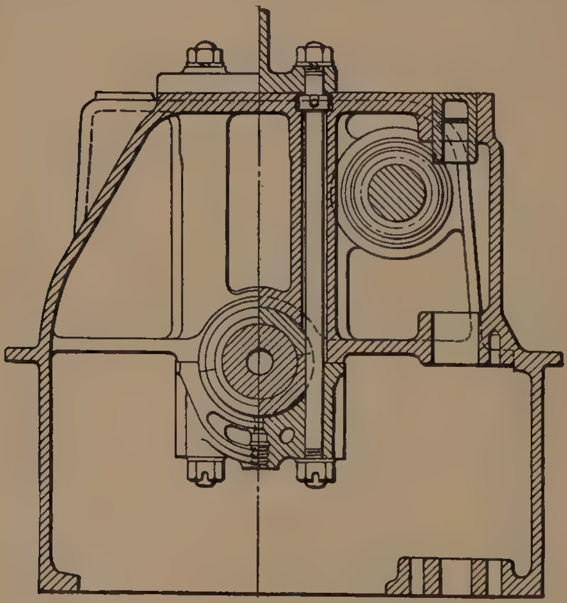


FIG. 49.—USE OF "THROUGH" BOLTS FOR HOLDING DOWN CYLINDER BLOCK.

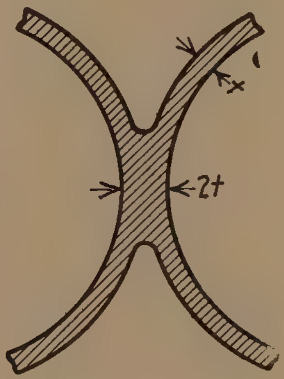


FIG. 50.—COMMON CYLINDER WALL.

In laying out the valve pocket, enough clearance must be allowed all around the valve seat so the gases will pass fairly uniformly through all sections of the valve port.

Core Hole Closures—Jacket cores must be accurately and securely supported, and core prints additional to those at the upper end of the block and at the water inlet must be provided. These prints leave holes which must be closed so as to be water-tight. It was formerly customary to tap them and screw in pipe plugs, but at present the so-called Welch patent plug is generally used for this purpose. This consists of a circular disc of sheet steel, which is dished in a press, whereby the cylindrical surface at its circumference is transformed into a conical surface, with a sharp edge on the convex side.

The hole to be closed, which generally passes through a boss formed on the jacket wall, is drilled out and counterbored to the exact diameter of the Welch "plug"; and the latter is inserted with its convex side toward the outside. The disc is then expanded by a direct blow with a hammer or by means of a flat or a very slightly concave punch. Its sharp edge is thereby caused to dig into the wall of the counter-bored hole, so that it is held firmly in place. A Welch plug is shown in position in Fig. 53 under the fan bracket.

Jacket Cover Plates—Blocks of six and eight cylinders frequently have a large opening cast in the jacket wall on the side opposite the valves, so that practically the whole side of the jacket is open. This helps to give a good support to the jacket core in casting and also facilitates removal of the core sand from the jacket. At the same time it reduces the weight of the completed engine, as the opening is closed by plates of sheet steel or sheet aluminum of considerably less thickness than the iron could be cast successfully. The cover plates are usually stamped in pan form, to give them the required stiffness, and sometimes a flange is turned up on the outer edge which stiffens the bolted rim and makes it possible to obtain a water-tight joint with a smaller number of cap screws. The joint is made with a brown paper or cork gasket, ¼ in. cap screws being placed approximately 1½ in. apart. Around the edge of the opening is cast a flange which is milled off to give a smooth seat for the gasket and also to provide a thread of sufficient length (¼ inch) to give the cap screws a good hold.

With long cylinder blocks it is something of a problem to ensure uniform circulation of the cooling water throughout the jacket. One solution consists in forming the water inlet on the jacket cover plate near one end and having a distributing pipe extend along the inside of the cover plate, with outlet holes in it directly opposite the water spaces between adjacent cylinder barrels. The tendency for the water to flow in greater quantity from the outlets near the entrance can be counter-acted by graduating the openings, making the more remote openings larger. Sometimes baffles are cast in the jacket to insure that all parts of the jacket receive an adequate supply of cooling water.

Water generally enters the cylinder jacket at the lowest point and leaves at the highest, so that the natural tendency of the water to rise as it becomes heated assists circulation. The water jacket must be so designed that no steam pockets can form; that is, there must be no part of the jacket from which the water cannot reach the outlet by a continuous upward course.

Inclined Valves—The large valve pocket required with L-head engines is a detriment from the point of view that it results in a compression chamber of shallow depth and large area in a horizontal plane, which presents a large cooling area for a given compression volume. To bring the valve seats somewhat closer to the cylinder and thus reduce the compression space wall area, some designers place the valves at a slight angle to the vertical, as shown in Fig. 51. With vertical valves the distance from the cylinder axis to the plane of the valve stems is determined chiefly by the consideration that the camshaft must be far enough from the crankshaft to be outside the sweep of the connecting rod heads. Another thing that influences this dimension is that jacket space must be allowed between the valve pocket and the cylinder barrel.

Siamesed Valves—Inlet valves in multiple cylinder castings are generally siamesed, that is, the spaces underneath the valve seats communicate with each other, and there is only a single outlet for the two valve chambers and only a single connection. There is some objection to siamesing the exhaust valves in this way, in the case of four-cylinder engines, because in these engines one cylinder begins to exhaust before the exhaust valve of the preceding cylinder closes, and if the two exhaust valves are siamesed it is inevitable that some of the exhaust gas from one cylinder will blow over into the other, thus preventing its complete evacuation. In six-cylinder engines the firing order can be so arranged that two adjacent cylinders will never fire in direct succession. The exhaust opening periods of the two cylinders will then be separated by 240°, which is more than the period of opening of one exhaust valve.

The valve spring chamber is usually closed by two cover plates. These are stamped from sheet steel and are dished so they will not deform materially even though the retaining nuts are drawn up quite tight in order to obtain considerable pressure on the gasket, so it will not leak oil. Generally the valve cover plates are held in place by studs which screw into bosses provided between adjacent cylinder barrels, but sometimes these studs screw into the valve tappet clusters which in turn are bolted against bosses provided on the cylinder block. The retaining nuts sometimes are of the form of a milled-rim hand wheel, so they can be removed without a wrench.

Outer Form of Cylinder Blocks—When twin and four cylinder blocks were still the rule for passenger-car engines it was the practice to use round outer surfaces to a large extent, joining the vertical to the top surface with a large radius and having the ends of the jacket hug the cylinder barrels. With the advent of six- and eight-cylinder blocks has come a preference for plane surfaces, and these blocks are generally made with a flat top and flat ends and with only a light rounding at the corners. Round surfaces, however, are being adhered to in Europe, particularly on the Continent, where the bosses for the cylinder-head bolts usually jut out prominently from the rounded contour of the head.

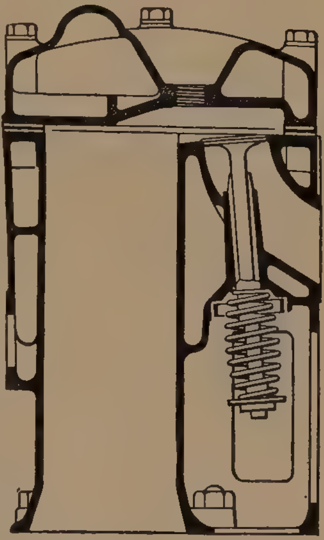


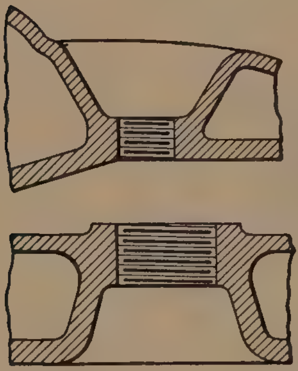
Fig. 51.—Valve Inclined to Cylinder Axis.

Length of Bore—In order that the piston may not wear a ridge in the cylinder bore at the ends of the stroke, it is desirable to have it over-run the ends of the bore slightly at both ends. Generally the engine is so designed that the top piston ring slightly over-runs the bore at the top end, but it is better to have it come to the end of the bore, as nearly as working limits permit. If the ring projected appreciably beyond the bore in the top dead center position, when the explosion occurred within the cylinder the pressure acting on the outside of the ring, would tend to compress it, which would make this ring ineffective and also put additional wear on the sides of the ring and the ring groove.

The total length of the finished bore evidently is equal to the length of stroke plus the length of the piston, minus the over-run of the piston at both ends. To facilitate getting the piston rings into the cylinder, the bore is chamfered at both ends.

Location of Spark Plug Bosses—In L-head cylinders the spark plug bosses are usually located over the passage between the valve chamber and the cylinder barrel. This is in accordance with the requirements that the spark should occur as near to the center of gravity of the mass of combustible charge as possible and that the spark points must not be in a direct line with the cylinder wall, where they would become fouled easily.

The length of thread of standard spark plugs is considerably less than the average thickness of the cylinder head, and to compensate for this difference a conical depression is formed in either the inner or outer wall of the head, as shown in Fig. 52. There appears to be one objection to each arrangement, which is probably the reason designers have not yet agreed on one or the other of these designs. If there is a depression in the outer wall, any water getting onto the top of the engine will collect in it and tend to cause rusting of the spark plug shell and its thread. A conical depression in the inner wall adds to the cooling surface of the combustion chamber. By using a special spark plug with extra length of thread the need for such depressions is eliminated.



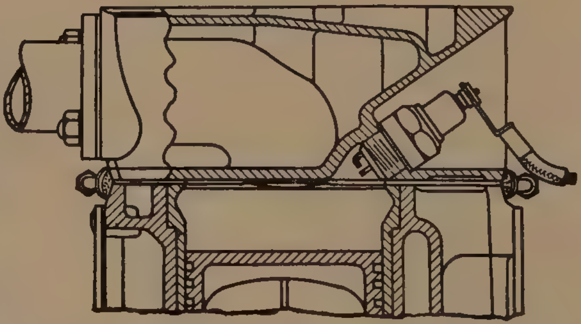


Fig. 52 (left).—Different Arrangements of Spark Plug Bosses.
Fig. 52A (right).—Spark Plug Boss in Valve-in-Head Engine.

In valve-in-head engines the depth of the head is altogether too great to permit of having the spark plug bosses extend through it vertically, and in the cylinders of such engines the spark plug boss usually extends through the head from the side at an angle, recesses being formed in the side of the head to obtain a square seat (Fig. 52 A). In the larger cylinders, of course, the vertical depth of the compression space is sufficient to allow of the plug being screwed into the compression chamber wall in a horizontal position.

In addition to the spark plug boss a priming cock boss must be provided in the head over each compression chamber. This is drilled and tapped for a ⅛-in. pipe thread.

Fan Support—In a considerable number of cars the stud for supporting the radiator fan is fixed to the engine cylinder, and if this arrangement is desired, provisions must be made in the cylinder design for the necessary screw holes and machined surface. The stud usually has quite a long overhang, and a rigid support must therefore be provided, with a relatively long vertical base. Fig. 53 shows a typical design. The stud is secured to the front of the cylinder block by means of four studs and nuts. On this stud is clamped the swinging arm carrying the shaft on which the fan pulley is mounted. In another design the cylinder block is provided with a cored hole at the front, which is bored out and tapped to a relatively large diameter and the fan stud, which is provided with a flange, is then screwed in.

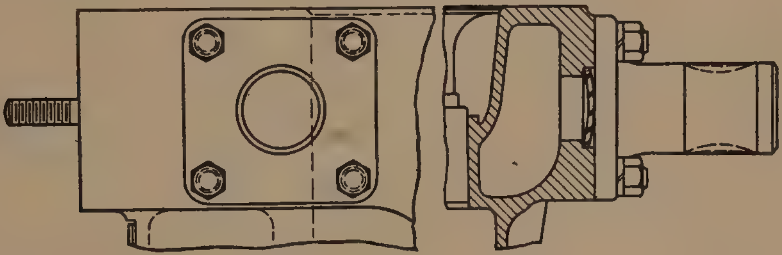
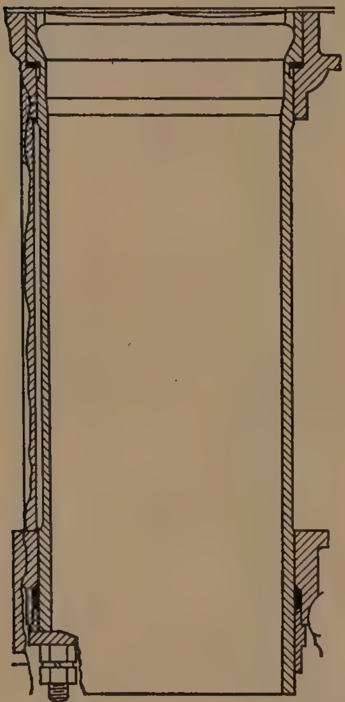


FIG. 53.—FAN BRACKET SUPPORT.

Aluminum Engines—In airplane engines, where lightness is paramount, the cylinders generally are made either of steel forgings or of aluminum castings, the latter receiving an insert or liner of steel, serving as a guideway for the piston. The most representative aluminum aircraft engine has been the Hispano-Suiza, in which both the cylinder wall and the liner are threaded, and the latter is screwed into the former. Aluminum cylinder blocks have been used also to a certain extent for automobile engines, in this country notably by Marmon and Premier. When Marmon first adopted aluminum for the cylinder casting, about 1916, both the outer and inner walls of the block were of aluminum. The inner wall was bored out, and a cast iron liner, $3/16$ inch thick, was inserted to serve as a guide for the piston. The liner was a slight shrink fit in the casting, the jacket being filled with hot water and the liner forced in cold. The aluminum casting comprised not only all of the cylinders, but also the whole of the crankcase except the oil pan. The valves were located in the cast iron cylinder head. After several years this form of construction was abandoned, the reason being, it is understood, that difficulty was experienced in keeping the cylinder bores round.

In a later design the cast iron liners, $\frac{5}{32}$ in. thick, were supported by the blocks only at belts about 1 in. wide, near the top and bottom, where the thickness of the liner was slightly increased (Fig. 54). At the upper end the liner had an outside flange, which entered a counterbore in the cylinder block. There was a similar counterbore in the lower guide for the liner. Packing material (asbestos and graphite) was inserted in this and compressed by a gland ring screwed into the counterbore. A gasket was placed under the flange at the upper end.



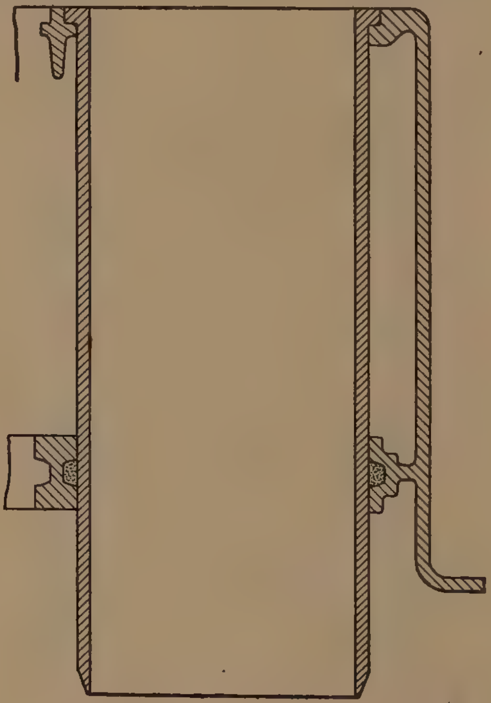


Fig. 54 (left).—Removable Liner for Aluminum Cylinder Block.

Fig. 55 (right).—Removable Liner for Tractor Engine-Block.

A similar construction was used in the Premier engine, except that at the lower end gaskets of cork and velumoid were placed between the end of the liner and a counterbore in the aluminum casting, while under the flange at the top end of the liner, only a velumoid gasket was used.

Aluminum engines have found considerable favor with German engineers in recent years, and at the Berlin automobile show in 1923 six such engines were exhibited. Germany during the War developed a large aluminum industry, and it is possible that surplus production of aluminum forced this development.

Removable Cylinder Liners—Cylinder liners as used with aluminum engines are also used to quite an extent in the cast iron engines of farm tractors, as the cylinder bores of such engines are very easily injured by dust in the air entering the carburetor and crankcase. In Fig. 55 is shown a construction with a rubber ring which is vulcanized to the liner.

Twin-Cylinder Block of Truck Engine—Sectional views of the twin-cylinder block of the Class B military truck engine are shown in Fig. 56 and outside views of the same block in Fig. 57. This is a four-cylinder engine with the L-head cylinders cast in pairs, the cylinder heads of each pair being a separate casting. When the cylinder heads are a separate casting the parting plane usually contains the top edge of the valve seats, and the pistons at the top end of their stroke come flush with the top end of the cylinder block. The compression space is therefore entirely in the cylinder head casting, most of it being provided by the valve pocket and valve passage.

Attention may be called to the liberal size of the water jacket in the Class B cylinder, particularly around the combustion chamber. The head casting is nicely rounded off, and the bosses for the holding-down studs are carried up high enough so they can be finished off in a surface milling machine. A housing for the valves is cast in the cylinder block.

This engine, which was designed for the Government by a committee of engine specialists during the War, is generally regarded as a very excellent type, and close copies of it are now being turned out by several manufacturers specializing on engines for heavy service.

Fig. 58 shows a plan view and two vertical half-sections of the cylinder head of the Class A military truck. The half-section on the left is taken through the cylinder axis and that on the right through the valve axis. There are two pairs of spark plug bosses, one pair over the passages from the valve pockets to the cylinders, the other pair in the cylinder axes. In addition there is a priming cock boss for each cylinder. The priming cock serves for injecting fuel into the cylinder by hand with a squirt can for starting, and also for injecting kerosene, etc., for cleaning the combustion chamber of carbon. Note the lugs cast onto opposite sides of the cylinder head for use in removing the head from the block. There is an oval boss on top of the head casting for the water outlet. All of

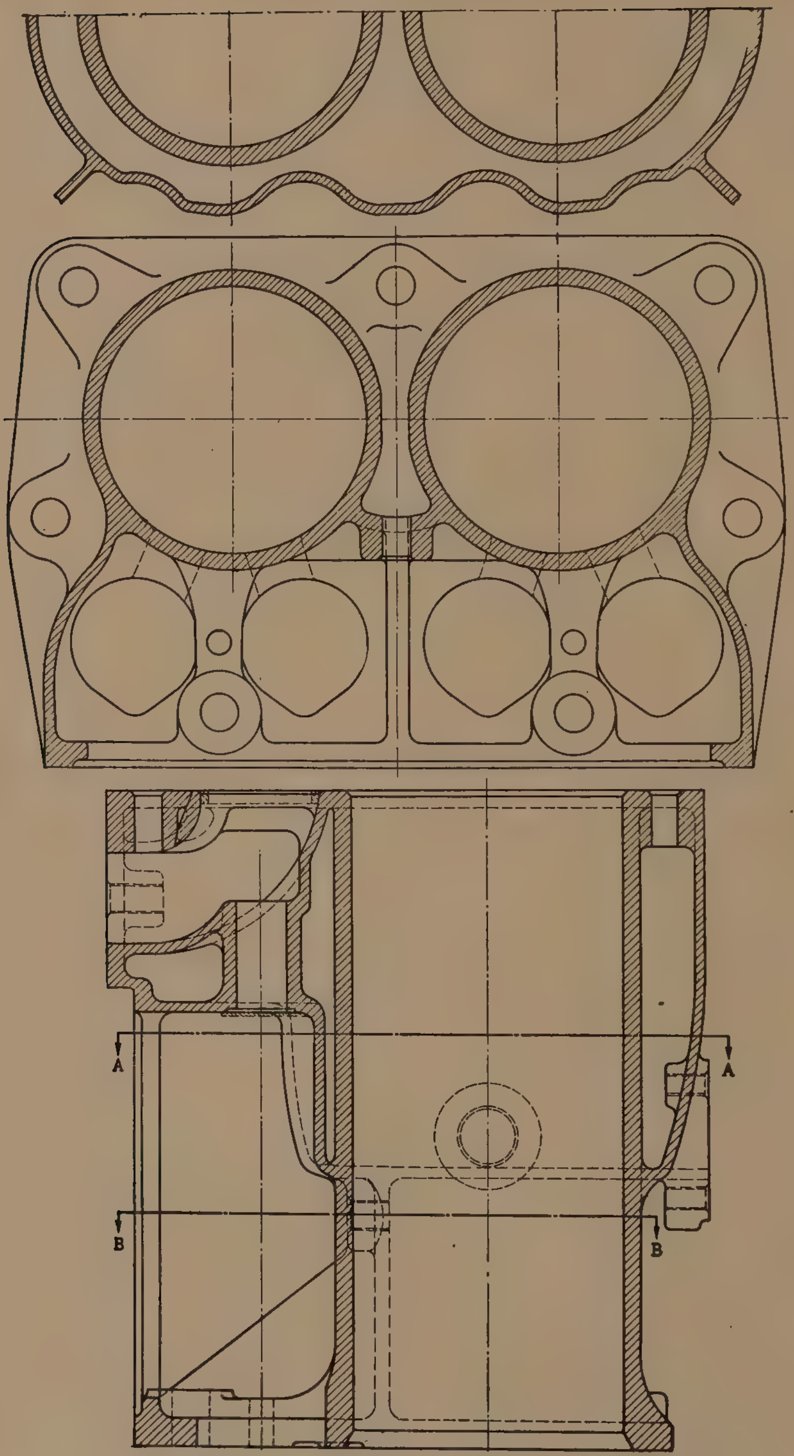


Fig. 56.—Sectional Views of Twin-Cylinder Block (Class B, U. S. A.)

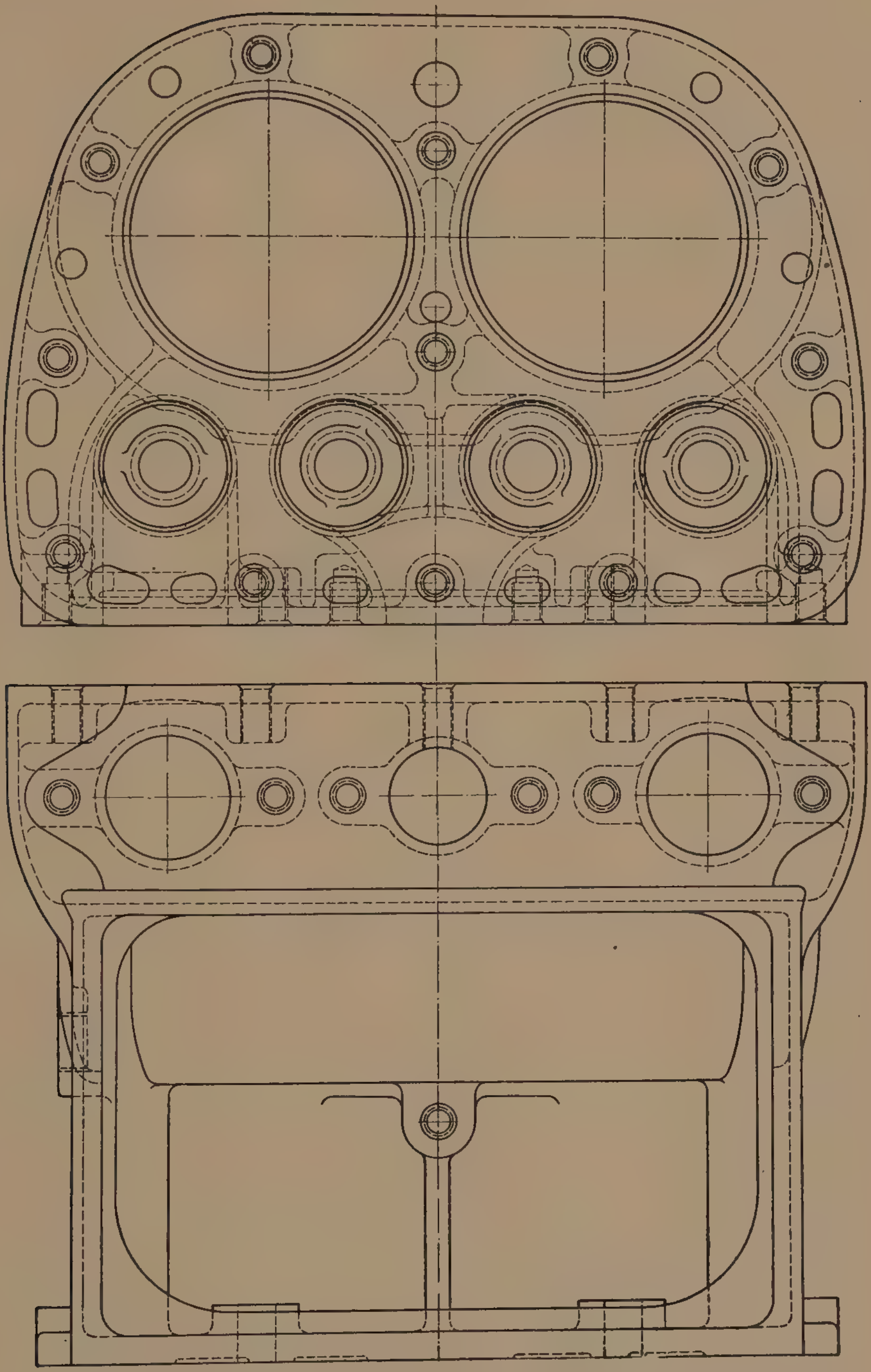


FIG. 57.—TOP AND SIDE VIEWS OF CLASS B CYLINDER BLOCK.

the seventeen bosses on the casting are finished off to a common level in one operation.

Cylinder Block with Integral Heads—In Figs. 59 and 60 (A and B) are shown a longitudinal elevation, partly in section, and two cross-sections of a six-cylinder block with integral head.

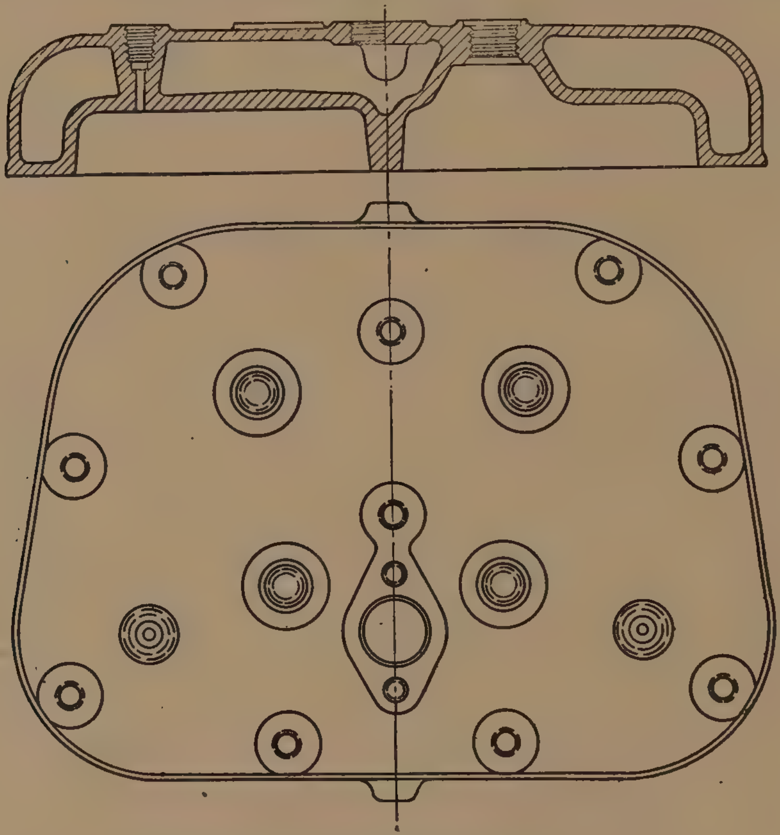


Fig. 58.—Removable Head (Class A, U. S. A.)

Although engines with integral cylinder head are practically obsolete so far as American practice is concerned, it was deemed well to show a design of this type to bring out certain features.

This engine is of the L-head type. It will be noted that the inlet valves of the three cylinders at one end are connected together by a passage cast in the block. The ports of these two internal inlet passages are connected by a manifold to a passage leading through the block between cylinders Nos. 3 and 4, to the opposite side, where the carburetor is bolted to the block. Each exhaust valve has an outlet port of its own. In this, as in many other engines, the inlet and outlet ports are on different levels, which is a very usual arrangement when more than two inlet valves communicate with each other through internal passages.

Placing the carburetor on the side of the engine opposite the valves was a popular practice before it became necessary to use exhaust heat for vaporizing the fuel. Sometimes the inlet passage was cored so as to extend around the front or the rear of the block, instead of straight across between cylinders, to reduce the length of the engine.

Attention may be called to the ribs between the cylinder head and the top part of the jacket wall. On top the jacket is left open to receive a water return fitting, which extends the whole length of the cylinder block and ends in front in a cylindrical outlet adapted to take a hose.

Combined Cylinder Block and Crankcase—Fig. 61 is a top view and Fig. 62 (A and B) are front and rear elevations of the cylinder and crankcase block of the Wisconsin Model Y engine. This is a valve-in-head engine of 3⅜ in. bore by 5 in. stroke. The cylinder barrels extend into the crankcase a distance of about 1 in. On both sides and at the ends of the block the water jacket is substantially ¾ in. deep. On the left-hand side there are cored passages through which the valve rods extend. Through these passages the valve chamber on top of the cylinder bead communicates with the crank chamber. One side of the crank chamber is nearly flat and is provided with large openings over which are bolted cover plates having the valve rod guide clusters formed integral with them. At the rear end bosses are provided on the crankcase portion of the casting for studs to fasten the bell housing to it. At the forward end a housing is cast on for the cam shaft and accessories drive gears. There is only one partition wall in the crankcase, that carrying the intermediate crankshaft and camshaft bearings. The crankcase is carried 2⅜ in. below the crankshaft axis, the object being to add to the rigidity of the block in the vertical plane. Strengthening ribs extend from the main bearing hubs to the bottom of the crankcase wall. There are three crankshaft main and three camshaft bearings, supported by the front, intermediate and rear cross walls, respectively. On the forward cross wall there is, in addition, a boss for an intermediate pinion in the pump and generator drive, and in the projecting flange at the forward end provision is made for mounting the pump bracket.

A vertical tunnel is cored in the block casting near the forward end to take a shaft for driving the ignition unit at its top and the oil pump at its lower end.

The flange of the front end drive housing is drilled and tapped for ⅜ in. cap screws at 3 in. centers. The bottom

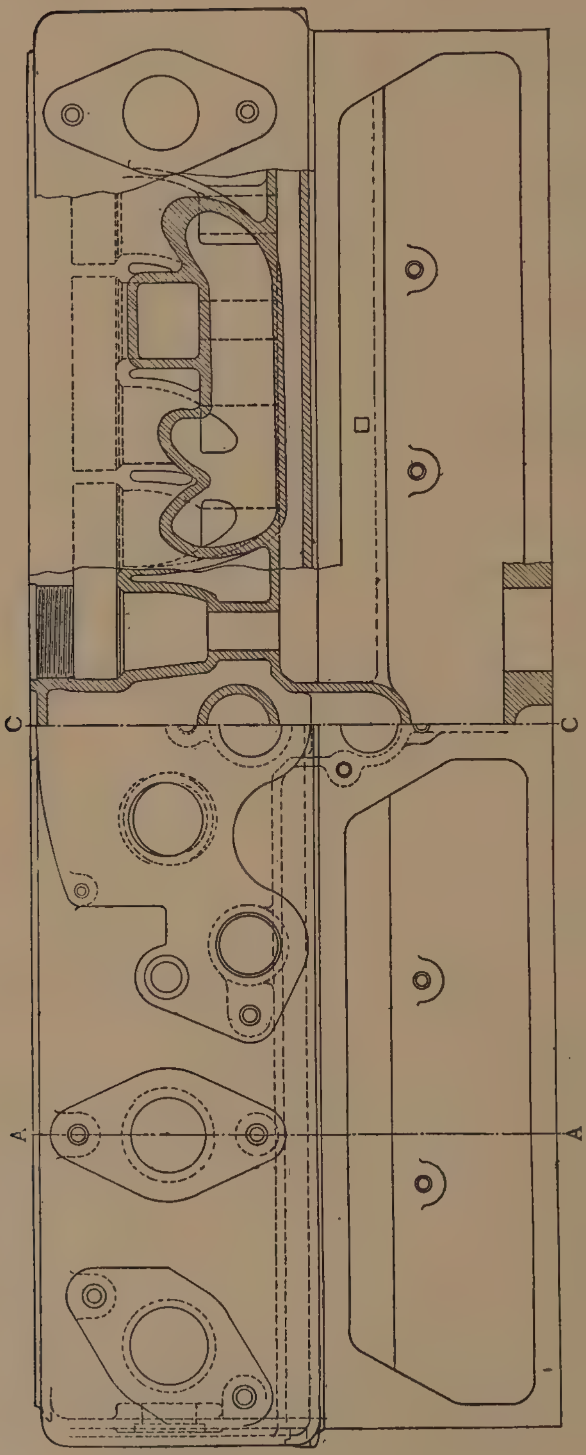
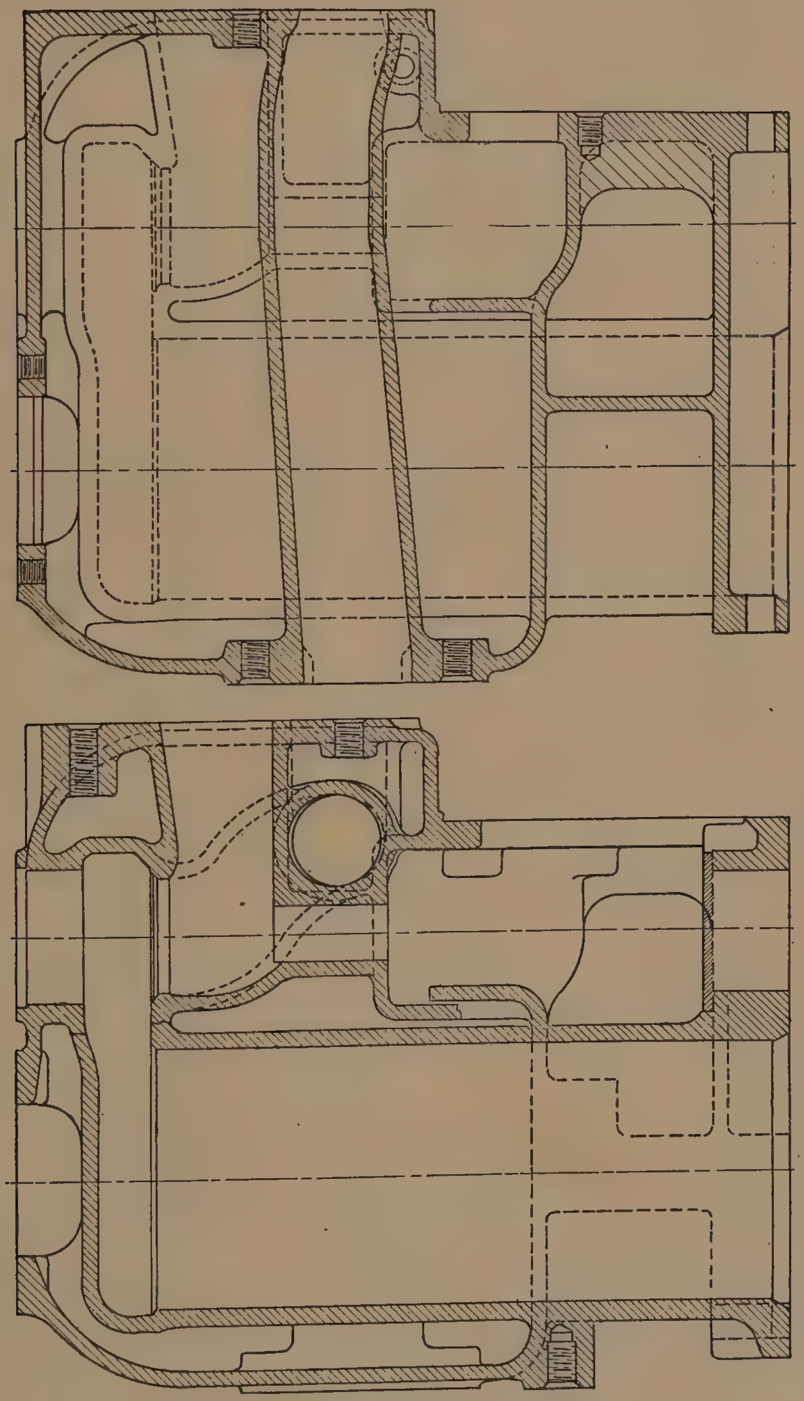


FIG. 59.—SIX-CYLINDER L-HEAD BLOCK CASTING WITH INTEGRAL CYLINDER HEAD (CONTINENTAL 9N).



B

A

FIGS. 60-A AND 60-B.—TWO CROSS SECTIONS OF CONTINENTAL BLOCK CASTING WITH INTEGRAL HEAD.

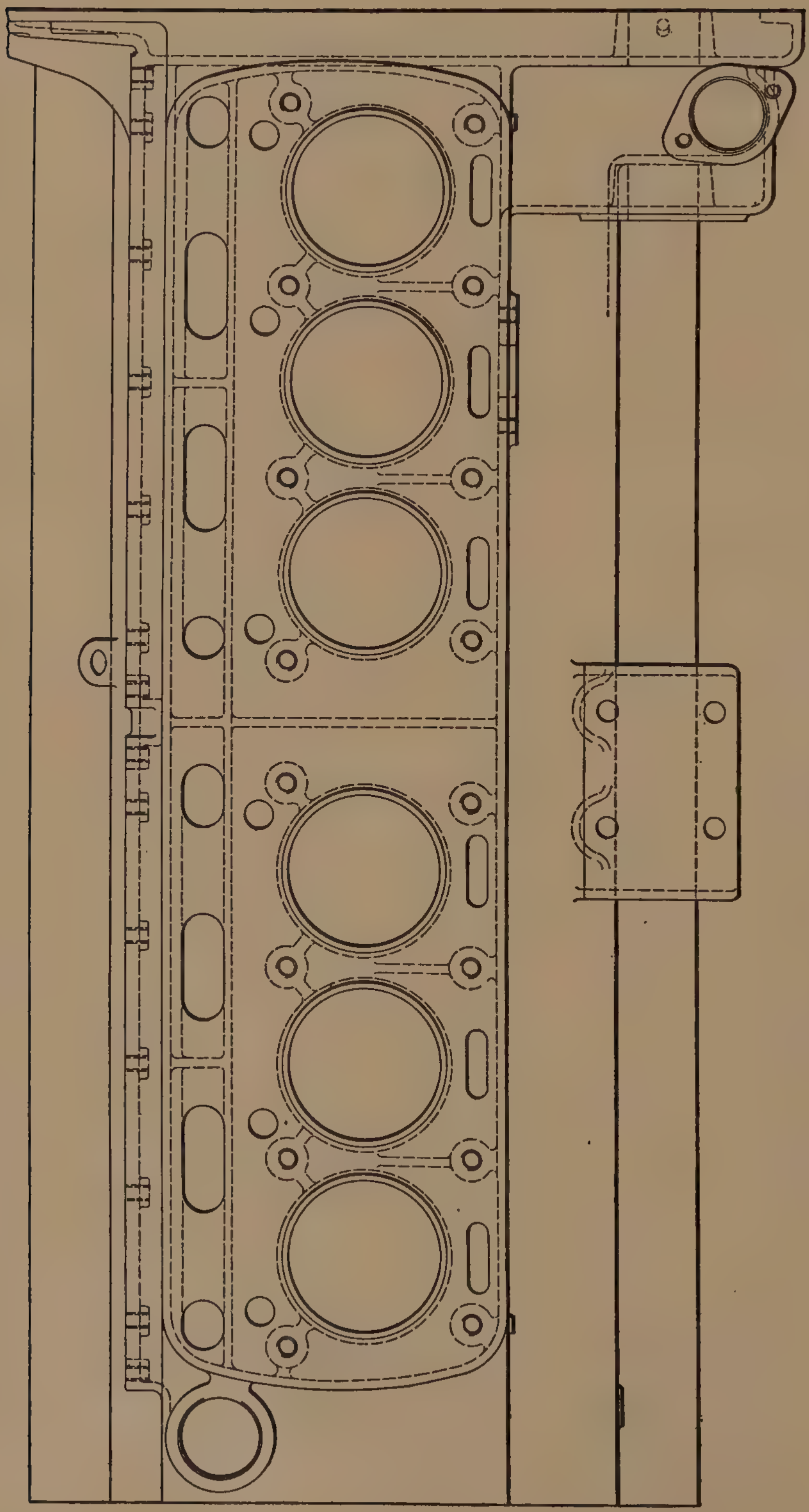


FIG. 61.—WISCONSIN CYLINDER-AND-CRANKCASE BLOCK, TOP VIEW.

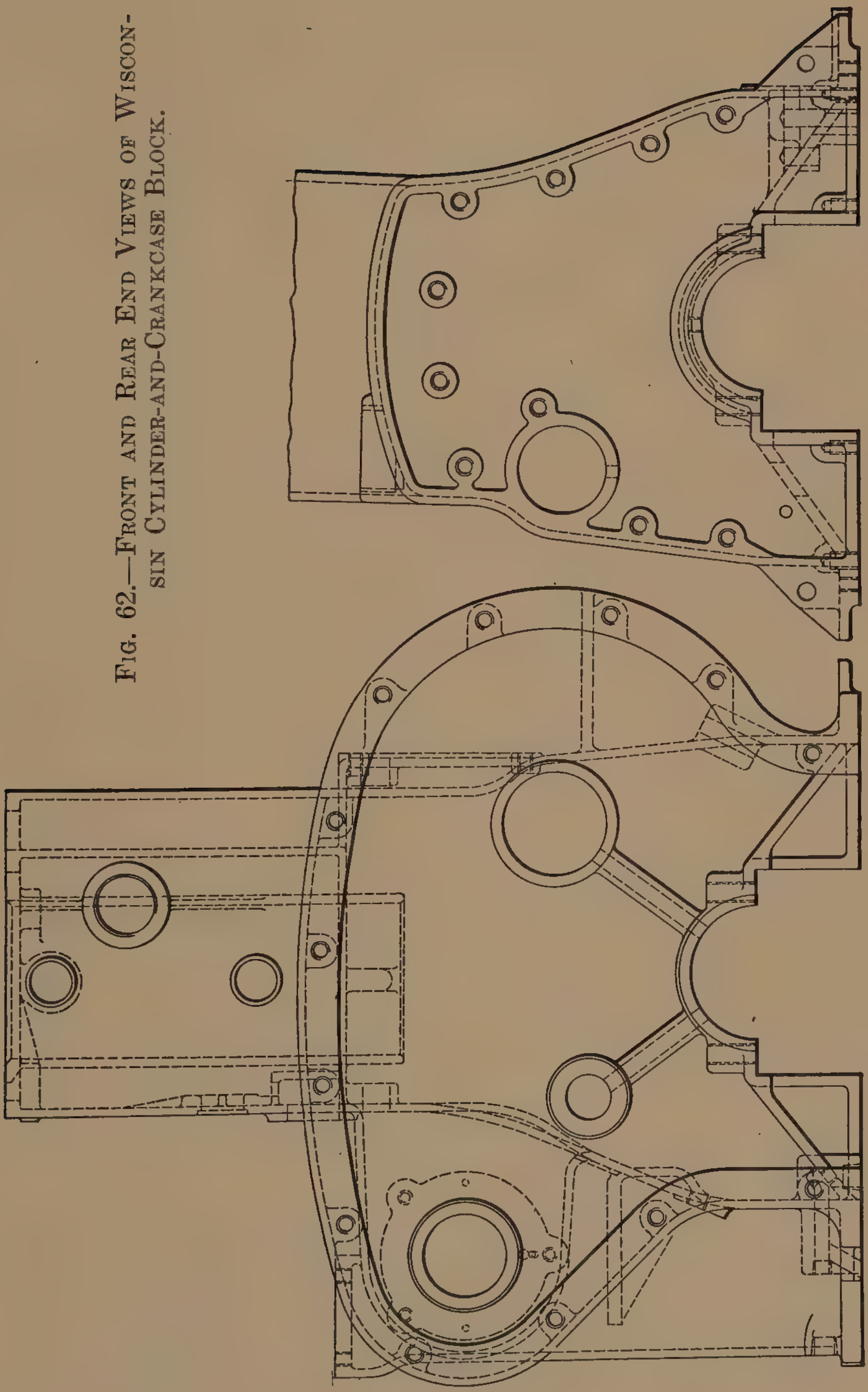


Fig. 62.—Front and Rear End Views of Wisconsin Cylinder-and-Crankcase Block.

flange of the block, which is unusually wide, is tapped for $\frac{5}{16}$ in. cap screws at $3\frac{1}{2}$ in. centers to hold the pressed steel pan in place. The finished cylinder wall is $\frac{7}{32}$ in. thick and the jacket wall $\frac{3}{16}$ in.

At the forward end of the cylinder block is cast a substantial boss which is tapped to take a shouldered stud to which the fan bracket is secured. The thread is $1\frac{3}{8}$–16 and is left-hand, so that the tension of the fan belt tends to screw the stud in. Altogether this cylinder-and-crankcase block is of very simple design and smooth or clean cut exterior.

Cylinder Head with Valves—In Figs. 63 and 64 are shown top and bottom views and sections of the cylinder head of the Marmon six-cylinder engine. The two views in Fig. 63 are looking down upon the cylinder head from the top and looking at it from the side of the manifolds, the latter view being partly in section. Of the views in Fig. 64, one is looking at the head from beneath, while the rest are transverse vertical sections in planes which are indicated in Fig. 63.

The inlet manifold is cast in the cylinder head, a passage of somewhat irregular shape, as shown in sections B-B, C-C, D-D, etc., extending practically the whole length of the block, and communicating with the individual inlet valves as shown in Section D-D. The single inlet to this passage is at the middle of the block and is clearly indicated in Section B-B. All of the exhaust valves have separate ports, as shown in Section F-F. In each cylinder the valve closest to the nearest end of the block is the exhaust valve. All of the seven valve ports are on the same level (see side view, Fig. 63).

There are sixteen bosses for the cylinder head bolts, and a section through two of these is shown in C-C. In Section E-E are shown four tapped holes. The two outside ones are for studs for the valve cover, while the two inner ones are for one of the rocker spindle supporting brackets. Six of these brackets and eight holding down studs are provided for. Close to the upper edge of the top view, Fig. 63, are seen the cored-in passages for the valve rods, four of the passages taking two rods each and the other four one each.

Between the valve ports there are bosses for studs which are used, together with yokes, to hold the manifolds in place, the yokes bearing on the flanges on the ends of the manifold branches. In Section B-B it will be noted that there is a drill hole through a boss into the inlet manifold, from the side of the head opposite the valve ports, which is intended for a suction connection to the vacuum tank. The hole in the top of the head at the very center is a core hole which is closed by a Welch plug. There are two other such core holes in the top, one in the plane M-M and the other symmetrical thereto on the opposite end of the head, and there are three in the bottom. The jacket in the head communicates with the jacket in the cylinder block through drilled holes which are located near the hold-down bolt holes. Drilling these water ports insures accurate registry and placing them near the hold-down bosses tends to give a water-tight joint with only moderate tension on the studs.

Pattern Making and Molding—In the design of the cylinder it is well to consult with the pattern maker, because a cylinder is at best a difficult piece to mold, and the advice of an experienced mechanic may obviate trouble later. Cylinder pattern making has become an art by itself, and it is a good plan to trust a job of this kind only to a man experienced in this particular line.

Cylinders must always be molded with the head downward, for the reasons that blow-holes, porous spots, etc., are most likely to occur near the top of the casting, and the head of the cylinder, which is the working end, must of necessity be of sound metal. In order that the metal may flow freely to every part of the mold, the latter must be thoroughly vented, and a high riser (opening on top which fills with metal after the mold is filled) is provided, so that the molten metal in the mold may be subjected to the pressure due to a considerable head, and all gases forced out through the vents. It is imprisoned gases that cause blow-holes and other imperfections.

Pickling Castings—When the castings have cooled they are cleaned of core sand, the seams, etc., are chipped off, and the castings are then pickled or cleaned of scale in a sulphuric acid solution. They are also carefully inspected for imperfections before any machine work is done upon them, so that no labor may be wasted on imperfect castings.

A ''continuous'' pickling machine is installed at the engine plant of Morris Motors, Ltd., in Coventry, England. It is in a single-story building, as illustrated in Fig. 65, and above the building there is a roller conveyor running parallel with the outside walls of the building, more than four times around, which serves as a storage track on which to keep a supply of engines to tide over any irregularity in deliveries from the foundry. Engine blocks are raised to the top of the track by an elevator of the tray type.

Each cylinder block gravitates to the pickling tank, virtually filling up the space vacated by the pickled engine that has just been dispatched. The block enters the pickling shop by way of a true helical roller track of smaller diameter, and

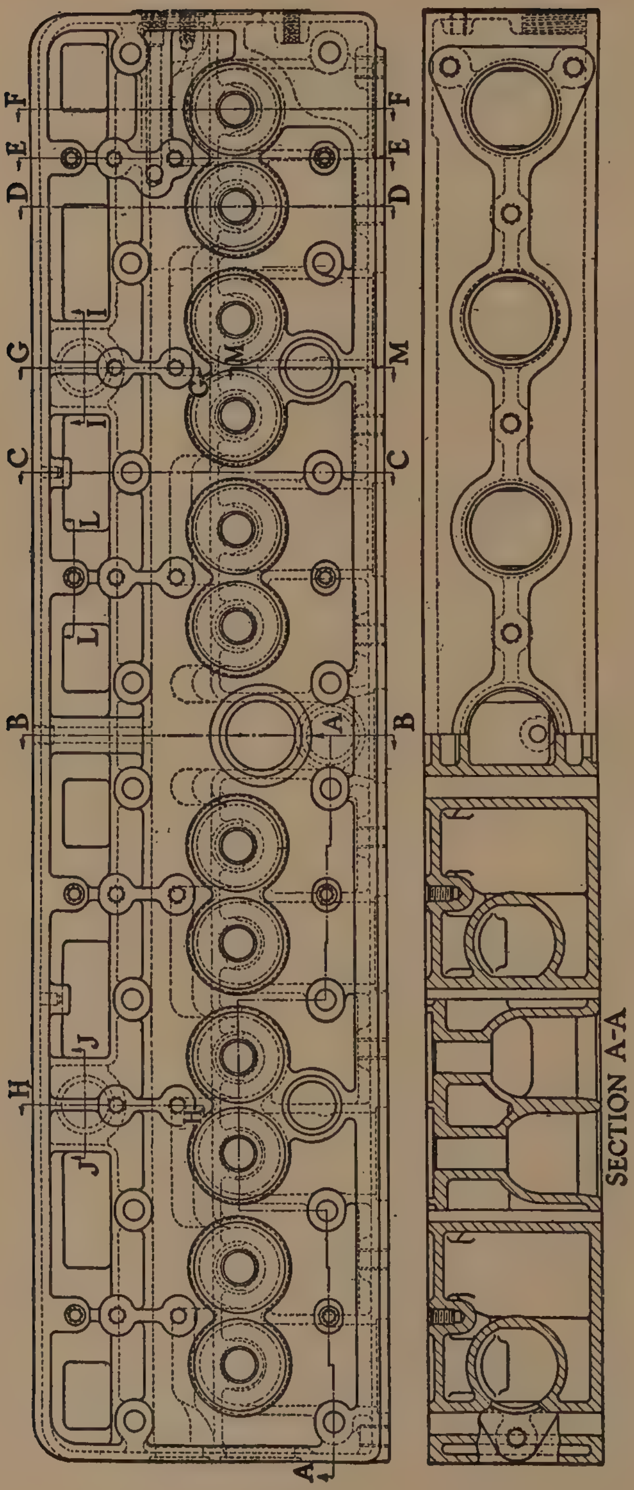


FIG. 63.—MARMON CYLINDER HEAD CONTAINING VALVES AND PASSAGES.

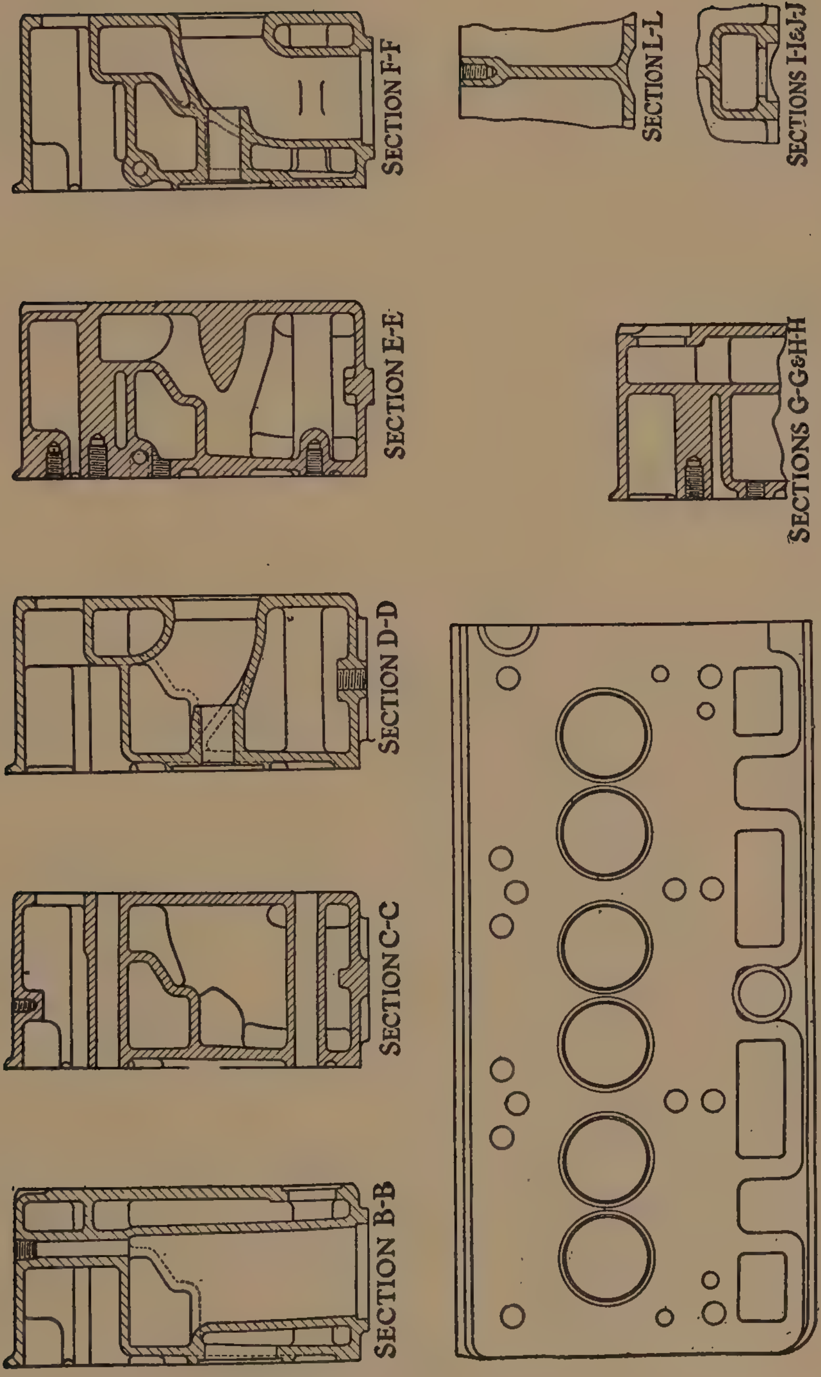


Fig. 64.—Detail Views of Marmon Cylinder Head.

is delivered onto a straight length of track for the final inspection and hardness test, and thence it is delivered to the pickling bath.

The latter consists of a lead-lined circular tank, 28 ft. in outside diameter, 4 ft. 6 in. wide and 3 ft. deep, which contains a dilute solution of sulphuric and hydrofluoric acid. The block is lifted by an air hoist and taken on a runway to what is referred to as the roundabout. This is an angle iron construction supported at the center by a vertical revolving shaft, with radiating arms which carry a circular rail 23 ft. 6 in. in diameter, from which the blocks are swung, two abreast, on grips made of mild steel and monel metal. The trip from the loading to the unloading station, that is, once around the bath, takes 2¾ hours, and this gives sufficient time to clean the block of sand and scale. The bath is kept at a moderate temperature by heating pipes, and the fumes are taken off by a fan in the roof of the building.

A somewhat similar pickling machine, known as the Mesta, is made in this country. There are five arms, arranged radially at equal angular distances and suspended from a rotatable frame. Each of the arms carries a heavy acid-proof tray suspended by rods, so that all of the trays can be raised and lowered simultaneously by means of a steam cylinder below the center of the rotating frame. In this way each tray is moved successively from the loading position to the various pickling tanks. The blocks on the tray remain in each tank for a period equal to that required to unload and load one of the trays.

When the loaded trays are immersed in the tanks, a semicircular hood fits down over the tank tops, collecting the fumes. When the arms are raised, collars on the suspension rods engage with the hood and raise it just sufficiently to permit it to revolve with the arms to the next position, where it is again lowered. The fumes are removed by means of a fan. Fan case and piping are of an acid-proof nickel alloy, while the shaft of the fan is of bronze and any spray that may collect at the bottom of the case is carried away through a lead drain pipe.

An alternate method of cleaning cylinder blocks is by a sand blast, but this is not used as much as the pickling bath, probably because it is not as effective in removing traces of core sand from the jacket.

Milling of Cylinder Blocks—The first machining operations on a cylinder block are usually the face-milling of the top and bottom surfaces. Such a facing operation on combined cylinder and crankcase blocks and on the corresponding

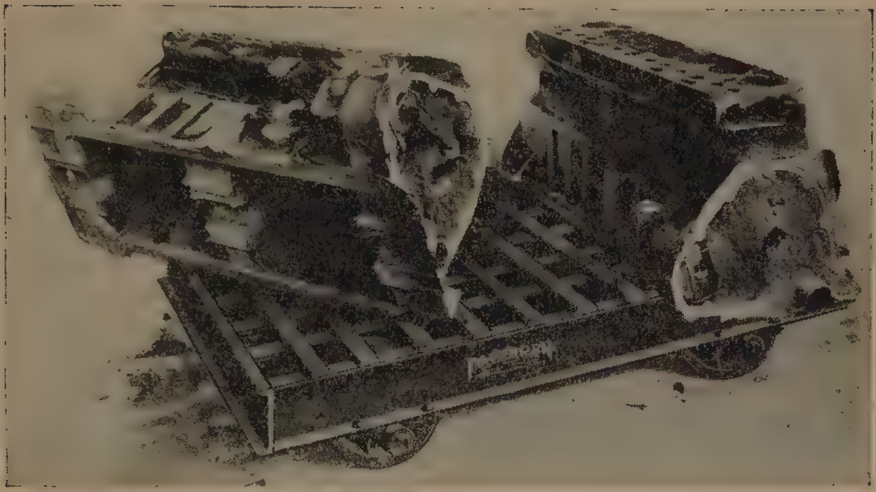


FIG. 64-A.—CYLINDER BLOCKS JUST OUT OF THE MOLD.

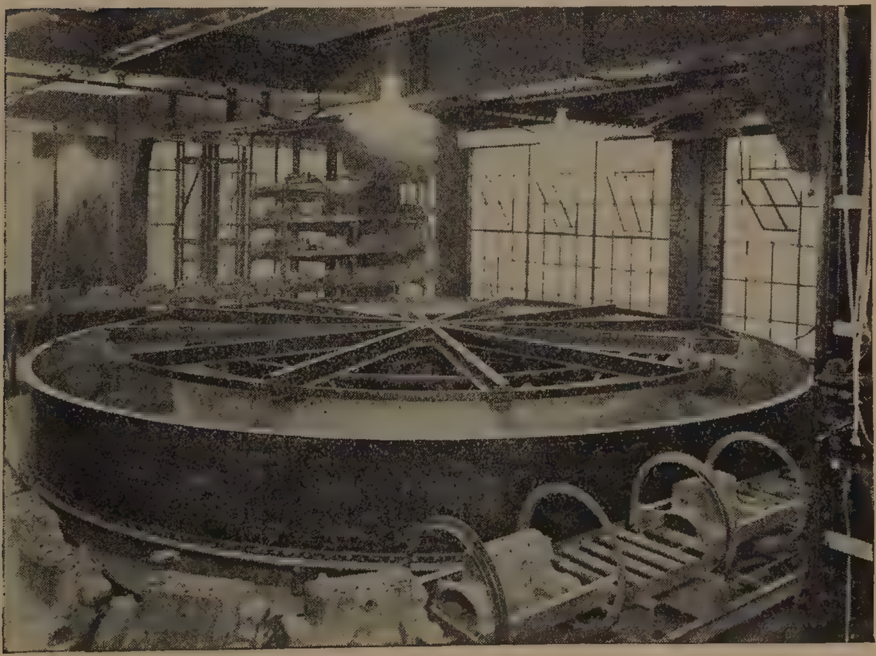


FIG. 65.—CONTINUOUS PICKLING MACHINE FOR CYLINDER BLOCKS (MORRIS).

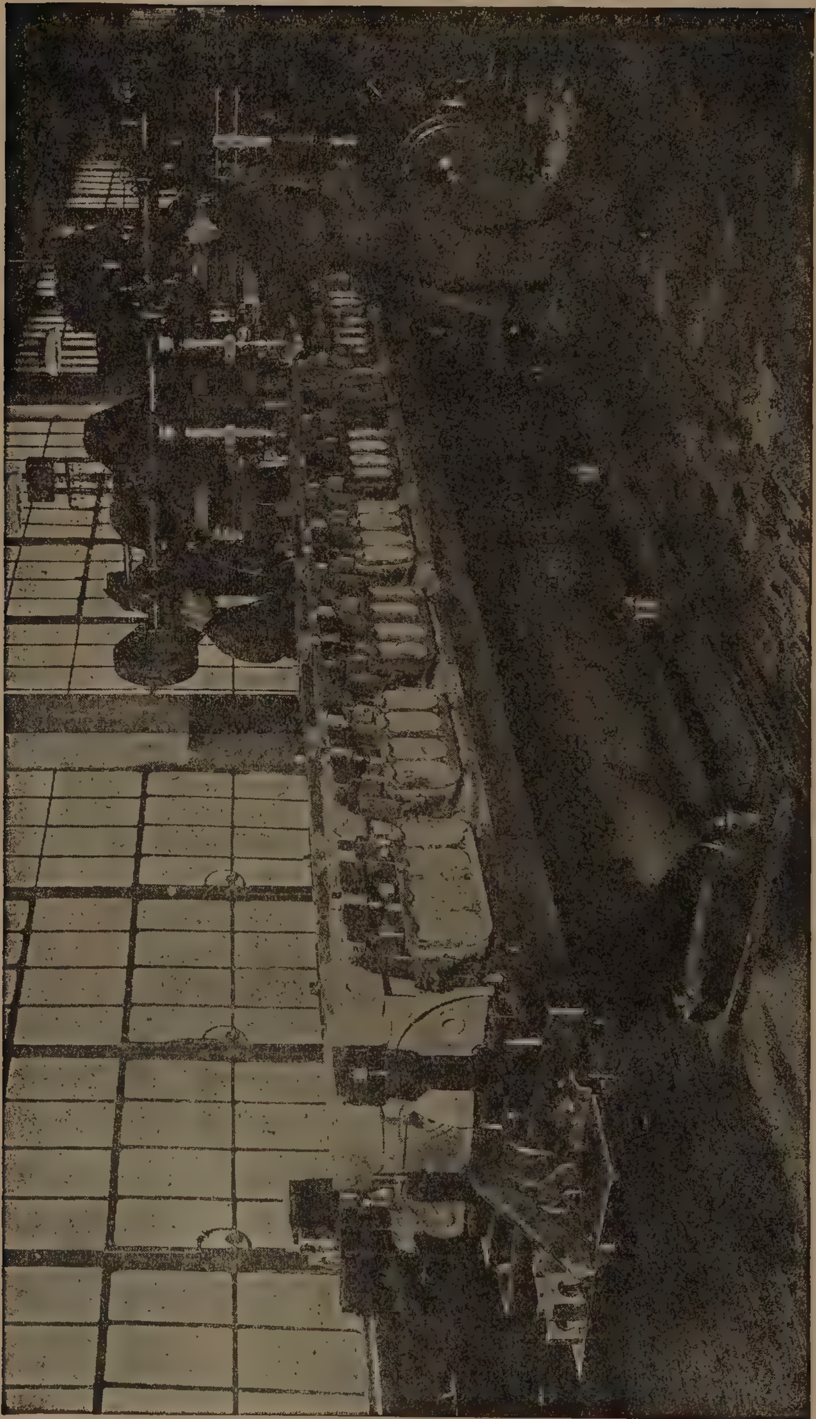


FIG. 66.—FACING CYLINDER AND CYLINDER-HEAD CASTINGS IN VERTICAL SPINDLE MILLING MACHINE.

heads, in a continuous milling machine of the table feed type, is illustrated in Fig. 66, the view being from the Dodge Brothers plant. A large number of cylinder blocks and cylinder heads are clamped to the table of the special four-spindle milling machine, the cylinders in two rows in the middle and the heads on both sides.

In the latest cylinder block "lines" or machinery layouts the top and bottom face of the block are usually milled in one operation, as in the Ingersoll rotary miller shown in Fig. 67. Two cuts are generally taken, a roughing and a finishing cut, of the order of $\frac{3}{32}$ and $\frac{1}{32}$ in., respectively. It is possible to hold the limits on the length to within 0.002 in. of each other. The machine shown holds six blocks and operates continuously, five blocks being worked upon while at one station a finished block is removed and a new one inserted.

Drilling Flanges—Next comes a drilling operation in which a multiple-spindle drill is used. In Fig. 68 is shown a four-way semi-automatic power feed drill in operation in the Dodge Brothers plant which drills the tap holes in the cylinder block after its top, bottom and end surfaces have been milled off. This machine drills fifty-five holes in one operation and works from the four sides of the casting simultaneously. The only attention given the machine is when the operator places the castings in position and removes them when completed. Fourteen holes are drilled in the top of the casting to take the studs that hold the cylinder head in place. Twenty holes are drilled in the bottom flange for the cap screws that hold the oil pan, and eleven and ten holes are drilled in the front and rear respectively, to take the cap screws that hold the front end drive cover and the transmission housing. The machine has a capacity of over 250 cylinder blocks per day. Working in conjunction with this is a twin machine with the same number of spindles. These are fitted with taps that cut the threads in the fifty-five holes in one operation.

More usually, however, the holes in the base flange are drilled alone in a multiple spindle drill, and in addition to the bolt holes, two dowel holes are drilled from which all subsequent machining operations on the block are located. For this drilling operation location is by V-blocks and stops entering the two outside cylinders.

Cylinder Boring—The next operation is the boring of the cylinders, which is done generally in a vertical boring machine with as many spindles as there are cylinders in the block. Foote-Burt cylinder boring machines are largely used for this operation. Fig. 69 shows a six-spindle cylinder boring ma-

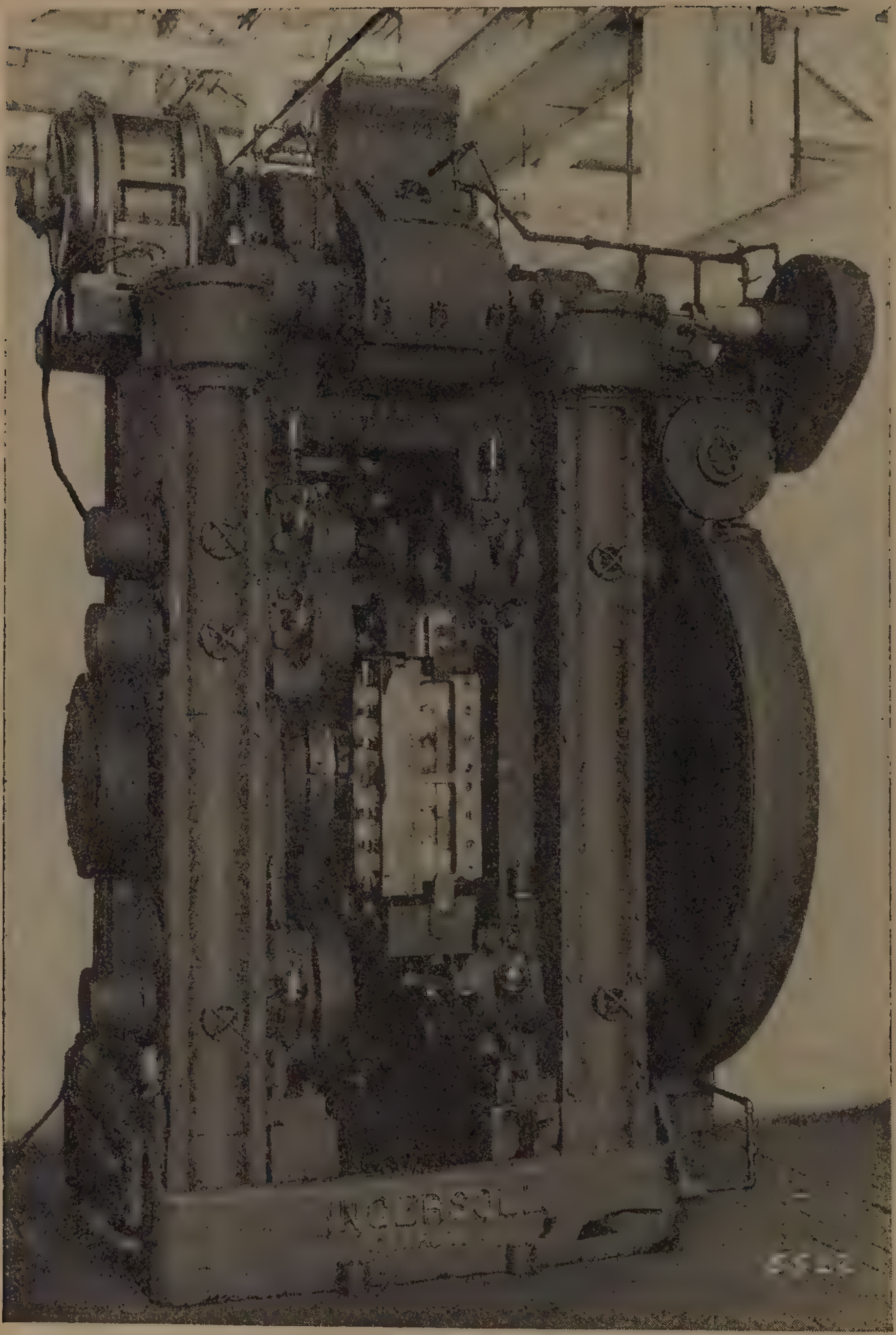


Fig. 67.—Facing Top and Bottom of Cylinder Block Simultaneously in Ingersoll Rotary Milling Machine.

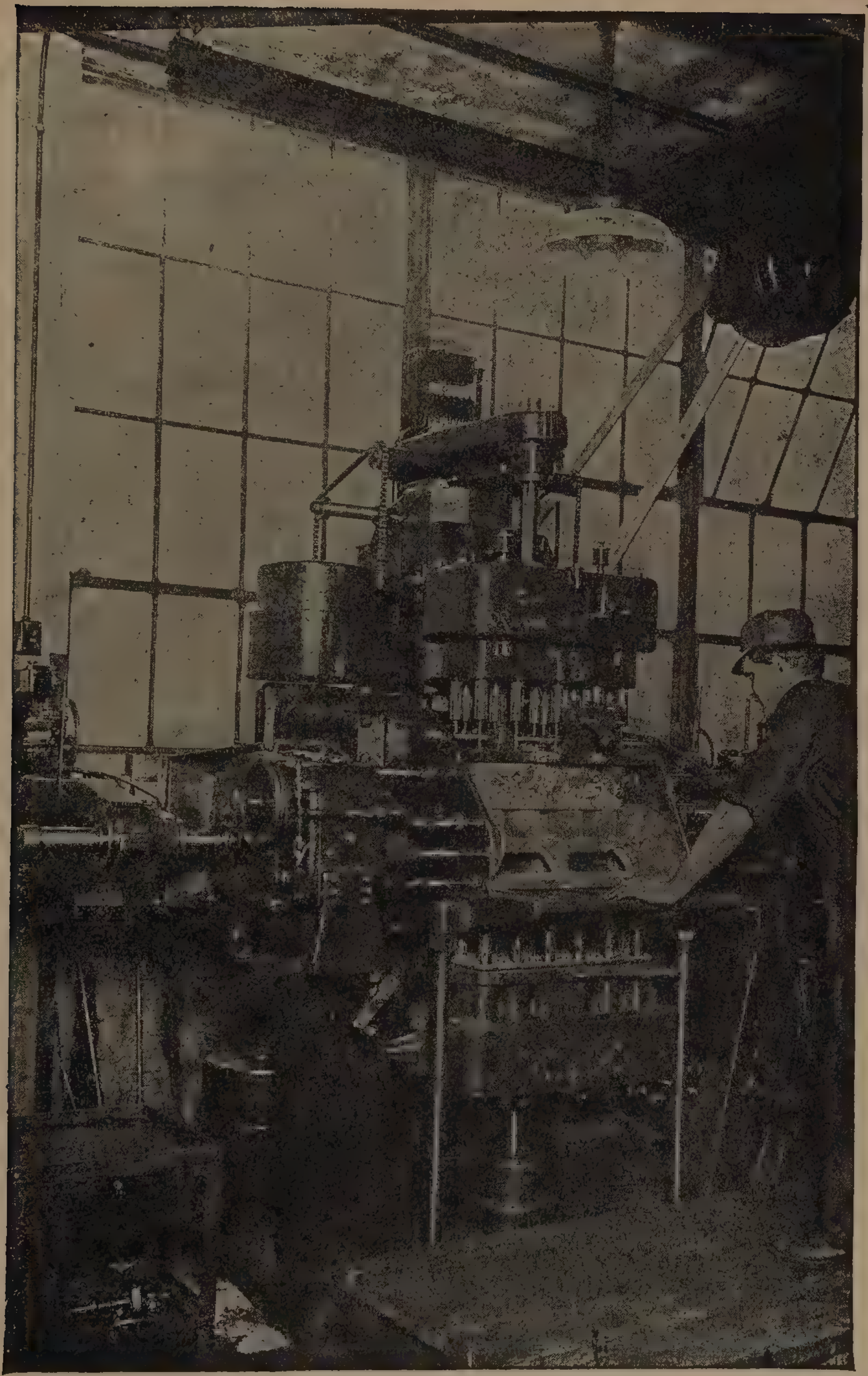


FIG. 68.—DRILLING CYLINDER BLOCK IN FOUR-WAY DRILL PRESS.

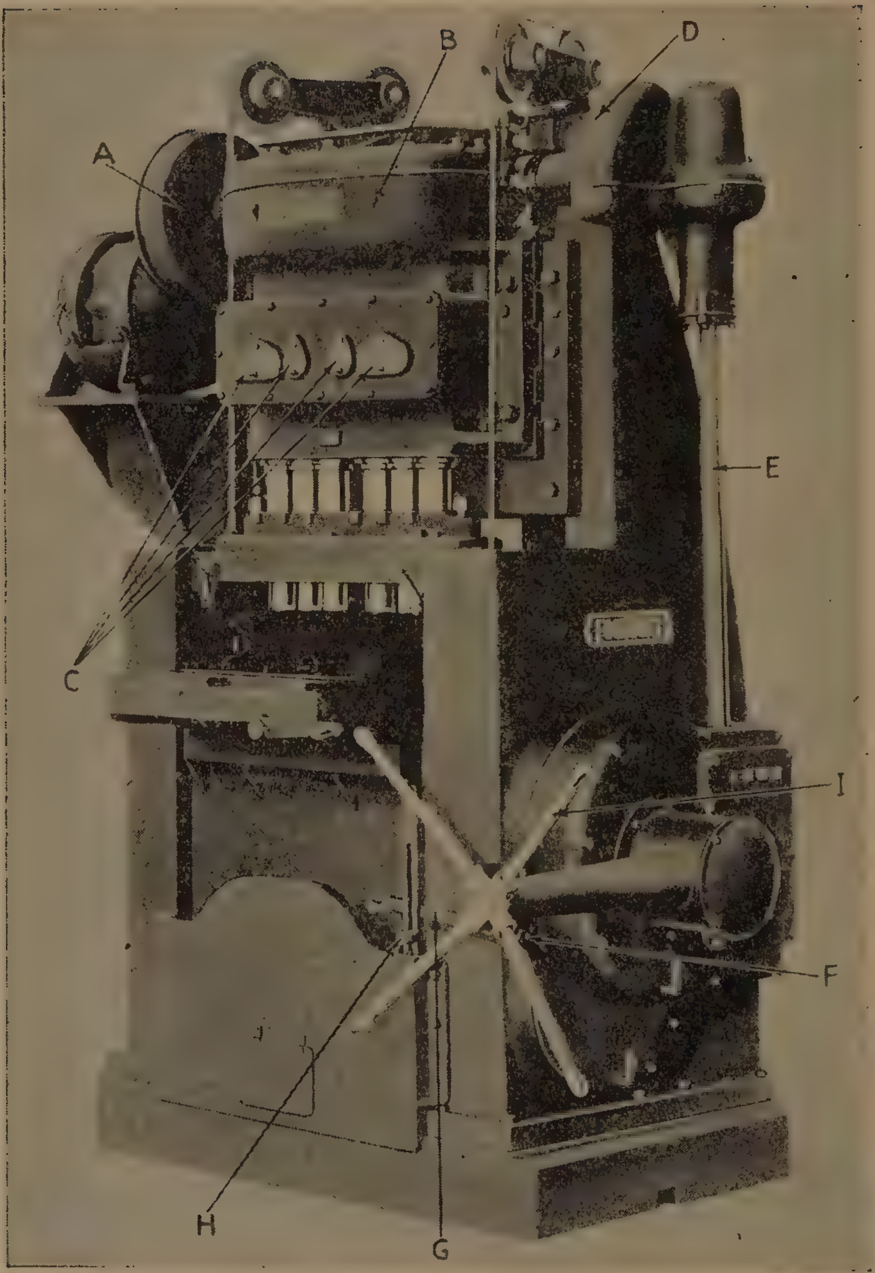


Fig. 69.—Six-Spindle Cylinder Boring Machine (Foote-Burt).

chine of this make. A horizontal shaft in the head carries two worms which mesh with a wormwheel each, located in the gearcase B. Each of these worm wheels is mounted at the top end of the middle one of three parallel vertical spindles. Under the worm wheel there is a spur gear which transmits motion to the other two spindles in the same group.

On this machine the head is stationary and the feed is accomplished by raising the work table. Four feeds are provided by means of two sets of pick-off gears in case D. From the main driving shaft in the head power is transmitted through the outside vertical shaft E to the feed mechanism for raising and lowering the table. By raising lever F a knock-out worm at the lower end of shaft E is engaged with the worm wheel and thereby the feed engaged. When lever F is raised it is latched at the upper end of bell crank G and when in that position it holds the worm in engagement with the wheel, resisting the pressure of a spring which tends to disengage the worm.

Secured to the work table is a slotted bar on which the feed trip may be set in any position. When the table reaches the end of the upward stroke trip H engages bell crank G and by rocking it causes the latch to release hand lever F.

The table is counterweighted. Steel cables connected with the table carry the usual counterweights which slide in cylindrical guards at the back of the machine.

In boring cylinders, two cuts are generally taken, the first or roughing cut increasing the diameter by about ⅛ in., while the second or finishing cut removes about $1/16$ in. on the bore. The cylinder block is located on the table of the boring machine by its reamed locating holes and is clamped down.

After the roughing cuts have been taken the cylinders frequently are annealed. This relieves internal stresses due to shrinkage, etc., and renders the metal of more nearly uniform hardness. The annealing process consists in slowly bringing the castings up to 1000° or 1050° Fahr., maintaining them at this temperature for about two hours and then allowing them to cool slowly in the furnace, the whole process occupying about ten hours. If the castings are not annealed, they are generally allowed to "age" or "season" for a period varying from a few days to a few weeks.

The Water Test—After the rough boring the block is subjected to a test for water-tightness under an hydraulic pressure of about 60 lbs. per sq. in. For this purpose a fixture is used similar to that shown in Fig. 70, which permits of rapidly connecting the water line to one opening of the jacket and closing up all other openings. This fixture consists of a skeleton frame designed to receive the cylinder casting. The frame is provided with two trunnions by means of which it is supported on the standard, Fig. 71, and which permit of tipping it to any angle, for examining the bore and all parts of the casting. The castings are held in place in the frame by means of three thumb screws, two pressing laterally against the lower unjacketed portions of the cylinders and the third centrally against the valve chamber. All holes into the jacket must be plugged, except the one through which the hydrostatic pressure is applied. The design of the fixture would naturally vary with the form of the casting to be tested. The one here shown was designed for a twin-cylinder casting with

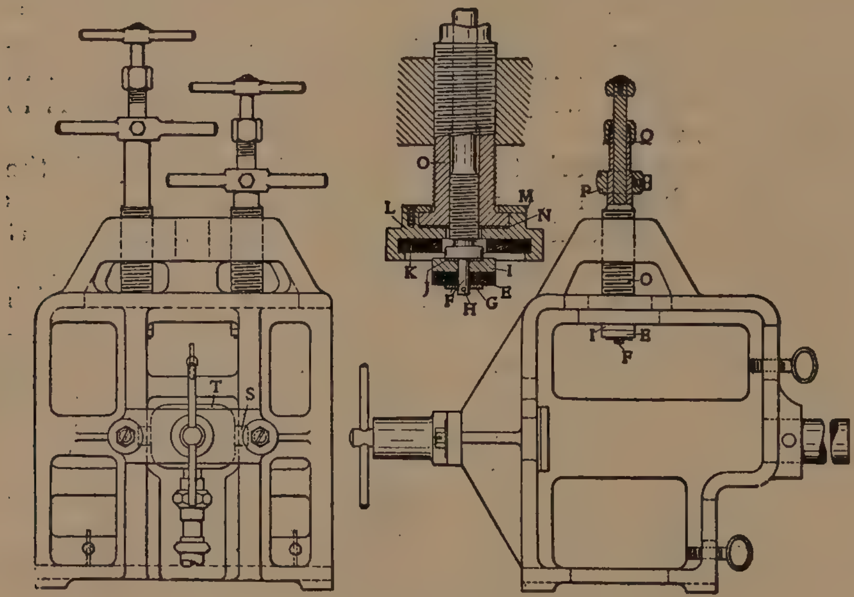


FIG. 70.—WATER-TEST FIXTURE.

central openings in the cylinder head and jacket head. The hole in the cylinder head is closed by means of a rubber gasket E which is held in place by a copper washer G and cotter pin H. This washer is backed up by a steel thrust disc I which has a copper washer J interposed between it and a shoulder on the spindle. The stem F is threaded at its lower end for applying pressure and is squared at its upper end for the operating handle.

The hole in the jacket head is stopped by a soft rubber plug K held in a thrust block L with a retaining flange M and copper washer N. This plug is operated by a hollow spindle O, which screws through the cast frame and carries the spindle F. Spindle O extends beyond band P to form a stuffing box, being threaded at its upper end to receive gland nut Q.

On the back of the cylinder casting for which this fixture was designed there is a large square hand hole. For plugging this hole and providing a water connection, a special construction is used. The trunnion at this side of the fixture is made of steel, bolted to the casting and drilled and tapped to receive a clamp screw. Grooves S are planed between the trunnion lugs, in which block T slides. This block is faced with a rubber packing which is held in position by flat-head screws. Block T is drilled and tapped to receive a pipe through which the testing water enters the jacket space. Connection to the water supply is made by means of a union with a short section of pipe and a length of rubber hose.

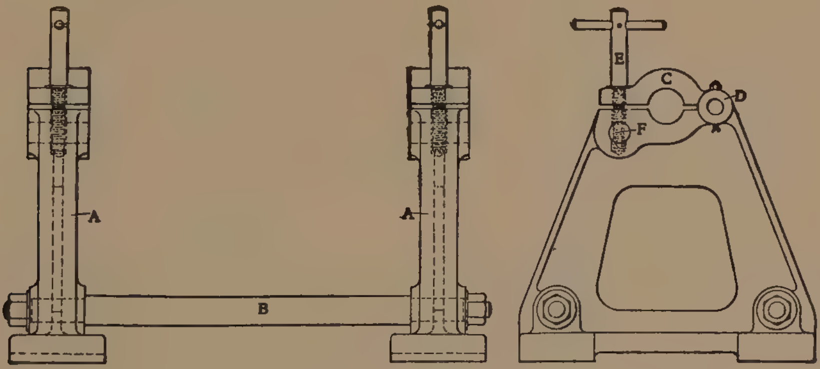


FIG. 71.—CYLINDER BLOCK TESTING STANDARD.

The standard in which the above fixture is mounted consists of two end castings AA held in place by spacers B. The trunnion bearings are provided with clamping caps C which are hinged at D and clamped with the T-handle E. The standard is made high enough to allow all parts of the testing fixture except the plug stems to pass over spacers B.

Finish of Bore—The final finishing of the bore of the cylinders is by either reaming, grinding, lapping, burnishing, or honing. The object is to obtain, as nearly as commercially possible, a perfectly cylindrical, mirror-like surface, the axis of which is absolutely perpendicular to the finished surface of the bottom flange, in which the pistons would move with comparatively little friction right from the start. The production of such a surface, however, involves considerable expense, and in the manufacture of engines for low-priced cars it is necessary to economize in every possible way, which to a certain extent accounts for the variety of finishing methods in use.

Reaming—The least expensive method of cylinder finishing, and which is in general use on engines of very large production, is the reaming process, which is performed in a machine similar to a cylinder boring machine, all of the cylinders of a block being reamed at once. It is not necessary to hold the block rigidly for this operation, as the cylinders already have been bored out and are substantially true. The allowance for reaming is about 0.080 in. on the diameter.

The original practice was to finish cylinders entirely by boring, four or five successive cuts being taken, the last with a broad-nosed tool to obtain a smooth surface. This, however, never gave entirely satisfactory results, the reason being that, owing to variations in the hardness of the cast iron and in the thickness of the wall, both the wall and the tool would yield more or less and an uneven surface result. This same difficulty obtains to a certain extent with reaming.

Lapping—Now, what is required in an engine cylinder is an absolutely cylindrical bore, which is the only kind that will permit of a gas-tight fit of the piston rings. To improve upon the original method the lapping process was introduced, which consists in working a lap, comprising a number of cylindrical sectors, up and down in the cylinder and at the same time rocking it back and forth. The sectors of the lap are held on the shaft or spindle in such a way that they can move radially outward, and light springs force them against the cylinder wall. A mixture of fine abrasive powder and oil is applied to the lap, and after the lapping process has been completed this is carefully removed by washing.

The compound reciprocating and rocking motion can be obtained in a number of different ways. In one instance the reciprocating motion is imparted to the lap spindle by a crank through a Scotch yoke, the connection between the yoke and the spindle being such as to allow the latter to revolve. The spindle is splined and passes through a spur pinion, also splined, which is held against endwise motion and is given a rocking motion by a horizontal rack meshing with it, the rack being reciprocated by another crank and Scotch yoke combination. This lapping process is being used at the present time to a certain extent.

Grinding—A great many cylinder bores are finished by grinding, special grinding machines for this purpose being produced by a number of makers. A grinding wheel of comparatively small diameter is used and is mounted on a spindle carried in a substantial rotating head. In the Heald cylinder grinder, illustrated in Fig. 72, the rotating head is carried in a combination of two eccentrics by means of which the wheel can be fed against the work and set for bores of different diameter. The axis of the spindle therefore is moved in a circle, in the same direction of rotation as the wheel itself. The grinder is provided with several changes of speed, a higher speed generally being used for the finishing passes.

Cylinder grinders are built only in single spindle form. In the case of multiple cylinder blocks, one bore is finished first and the table is then moved sideways a distance equal to that between cylinder axes, bringing the next bore into line with the grinder head, which can be accurately done by means of a micrometer adjustment. From 0.010 to 0.015 in. of stock on the diameter is usually allowed for grinding.

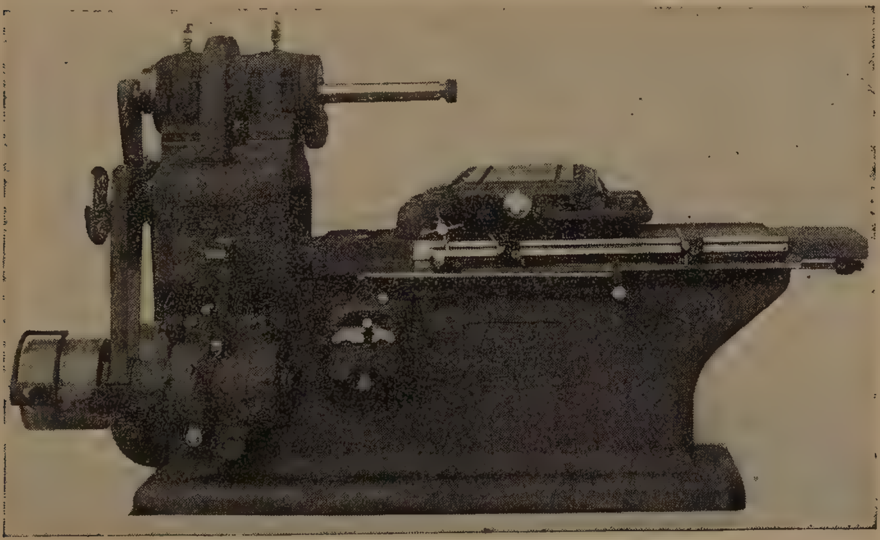


Fig. 72.—Heald Cylinder Grinder.

Cylinder blocks are mounted on the machine by means of an angle bracket type of jig, which has one plane surface that is fastened down to the table of the grinder, and another at right angles thereto to which the base flange of the block is clamped.

The abrasive and cast iron dust removed from the bore by the grinding operation are carried away by a suction pipe provided with a funnel which is placed over the top opening of the cylinder. In some shops water at about 180° Fahr. is kept circulating through the jackets while grinding, the object being to have the cylinder block at as nearly operating con-

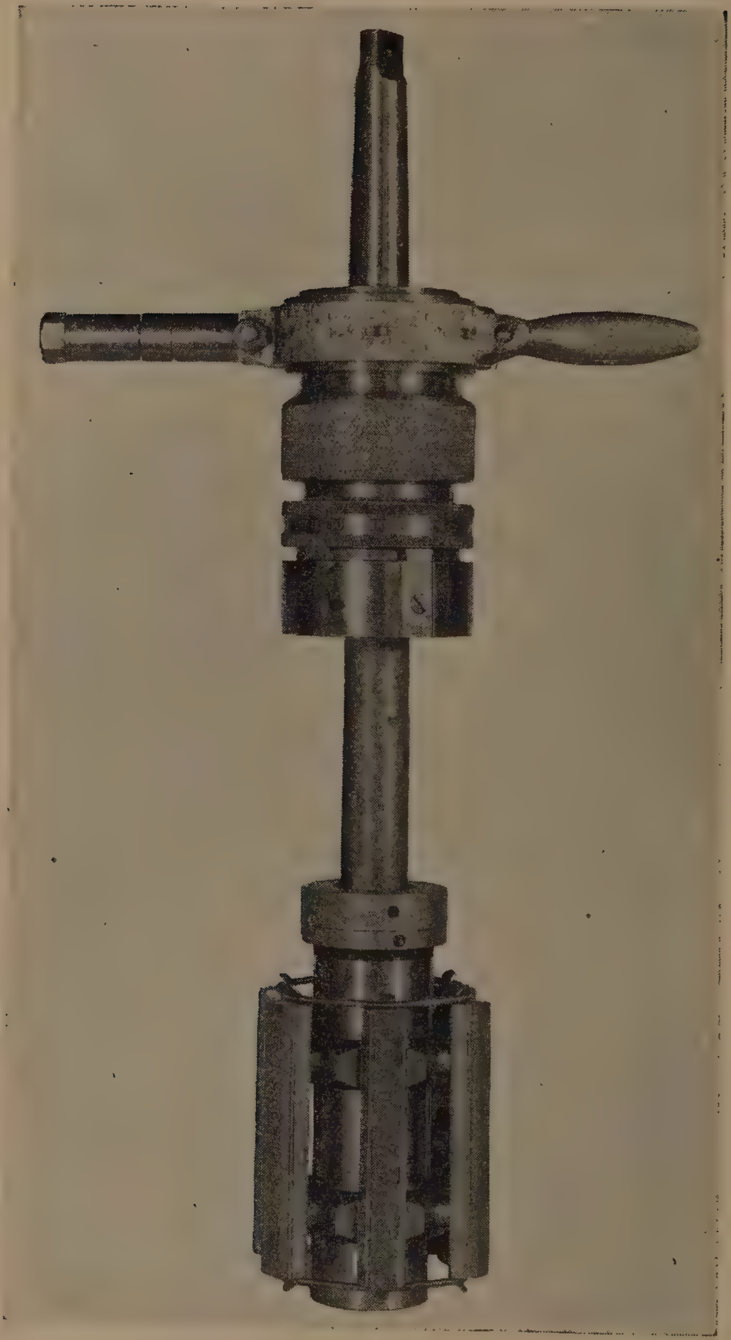


Fig. 73.—Hutto Cylinder Grinder.

(The hollow shaft connecting the upper and lower parts has "universal" connections, so the grinding head will be guided by the bore.)

ditions as possible. It is claimed that if the block is ground while cold and absolutely perfect bores are obtained, they will not be perfectly true when the cylinder is heated up. One objection to grinding hot is that if the bore is perfectly true while hot it will not be when cold, and a perfectly ground cylinder therefore may be rejected by the inspector. For this reason, when grinding hot it is best to inspect the cylinders while they are still hot. Cylinder bores are generally tested for accuracy by means of limit gages, of the type known as plug gages, the two ends of which differ in diameter by 0.001 or whatever the tolerance may be. One end should enter the bore, the other should not.

One advantage claimed for grinding over lapping and burnishing combined with reaming is that it gives better assurance that the cylinder axis will be perpendicular to the base surface.

Honing—A hone consists of a metallic frame supporting three or four abrasive stones which are pressed against the cylinder wall by springs. The hone is usually given a combined rotary and reciprocating motion. A hand-operated type is made which is revolved in the cylinder by a portable electric drill and at the same time is moved up and down. The hone is not used to remove any appreciable amount of metal but rather to eliminate the tool marks or slight roughnesses left by other methods of finishing.

Fig. 73 shows the Hutto grinder which embodies some of the features of the cylinder hone. Like the hone, it comprises a number of abrasive stones which are held in a fixture that is revolved inside the cylinder. However, whereas in the conventional hone the stones are pressed against the cylinder wall by spring force, in the Hutto grinder they are positively adjusted by means of a mechanism in the upper driving head which connects to adjusting cones in the cutter head through a square shank extending down inside the hollow floating (or universal) drive shaft. In the ordinary hone, on the other hand, the stones are forced against the cylinder wall by a coiled spring concentric with the cutter head, which presses against cones that engage conical surfaces on the holders for the abrasive stones.

An alternate method of finishing which is intended to give about the same result as honing is that of burnishing or rolling. This is accomplished with a tool carrying a number of hardened rollers, with a slight taper. The tool is revolved and at the same time slowly forced down into the bore. In one case the tool is made with a taper of 3°. Such a burnishing tool, which is used in finishing the cylinders for Mack truck engines, is shown in Fig. 74. The rollers are made of high-speed steel; they are guided by means of a cage as shown and are supported by a collar of hardened tool steel ground to size. For a 4¼-in. bore cylinder the rollers are made of 0.375 in. diameter, plus or minus 0.0001, while the supporting

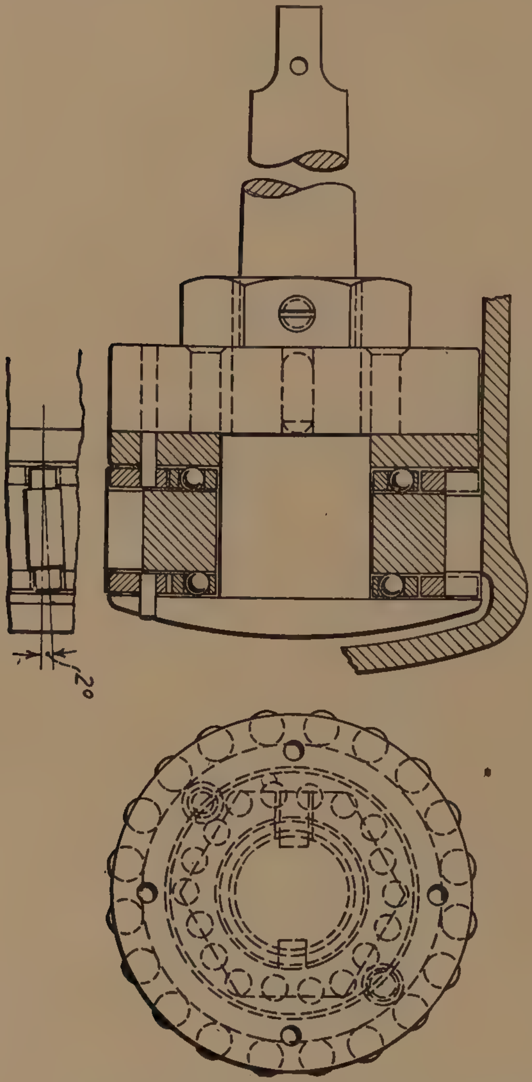


Fig. 74.—Cylinder Bore Burnishing Tool.

collars are made in nine sizes varying in diameter from 3.5035 to 3.509 in.

Both the honing and burnishing processes are generally preceded by reaming. One manufacturer, for instance, after boring, rough-reams the bores, taking out $1/16$ in. on the diameter, then finish-reams them, taking out 0.020 in., and finally burnishes them to take out the slight roughnesses left by the reamer.

According to a survey made early in 1924 by the S. A. E. Research Department, out of 24 engine builders from whom information was obtained, 12 honed their cylinders, 9 finish-ground them and 3 used the lapping process.

Surface Grinding—The same as the top and bottom surfaces, both ends of the block are milled off in a continuous milling machine. All these surfaces are sometimes finished in a cupped wheel surface grinder, which gives a truer and smoother surface than the milling cutters, and, therefore, gives a better assurance of a water or oil-tight joint. The operation of grinding the surface of cylinder blocks in a Blanchard surface grinder is illustrated in Fig. 75.

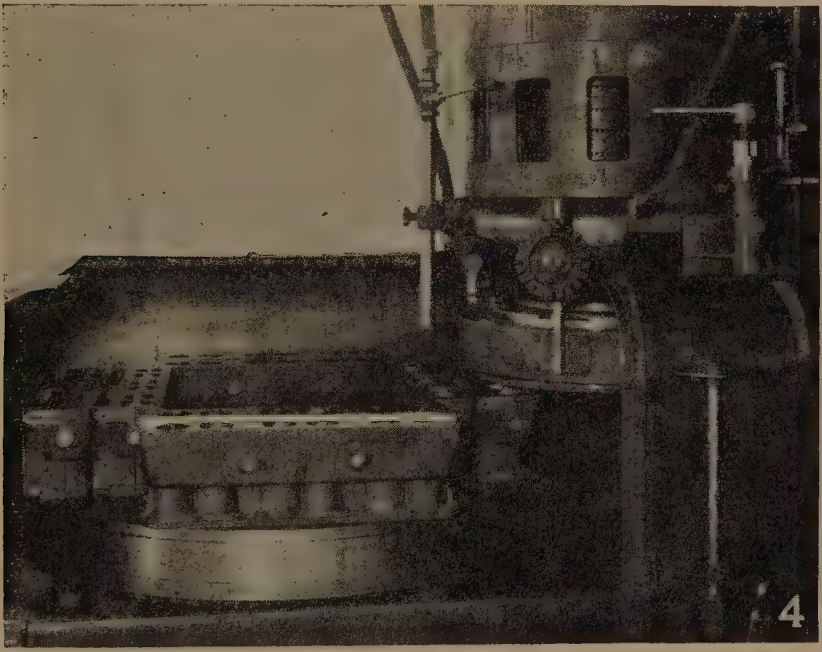


Fig. 75.—Grinding Top of Cylinder Block in Blanchard Surface Grinder.

Railroad Gang Drill—It is not intended here to go into the minor machining operations such as drilling and tapping the manifold bolt holes, counter-boring the valve ports, etc. Reference should be made, however, to the operations on the valve seats, valve guides, etc. The operations are sometimes performed in a railroad gang drill, as illustrated in

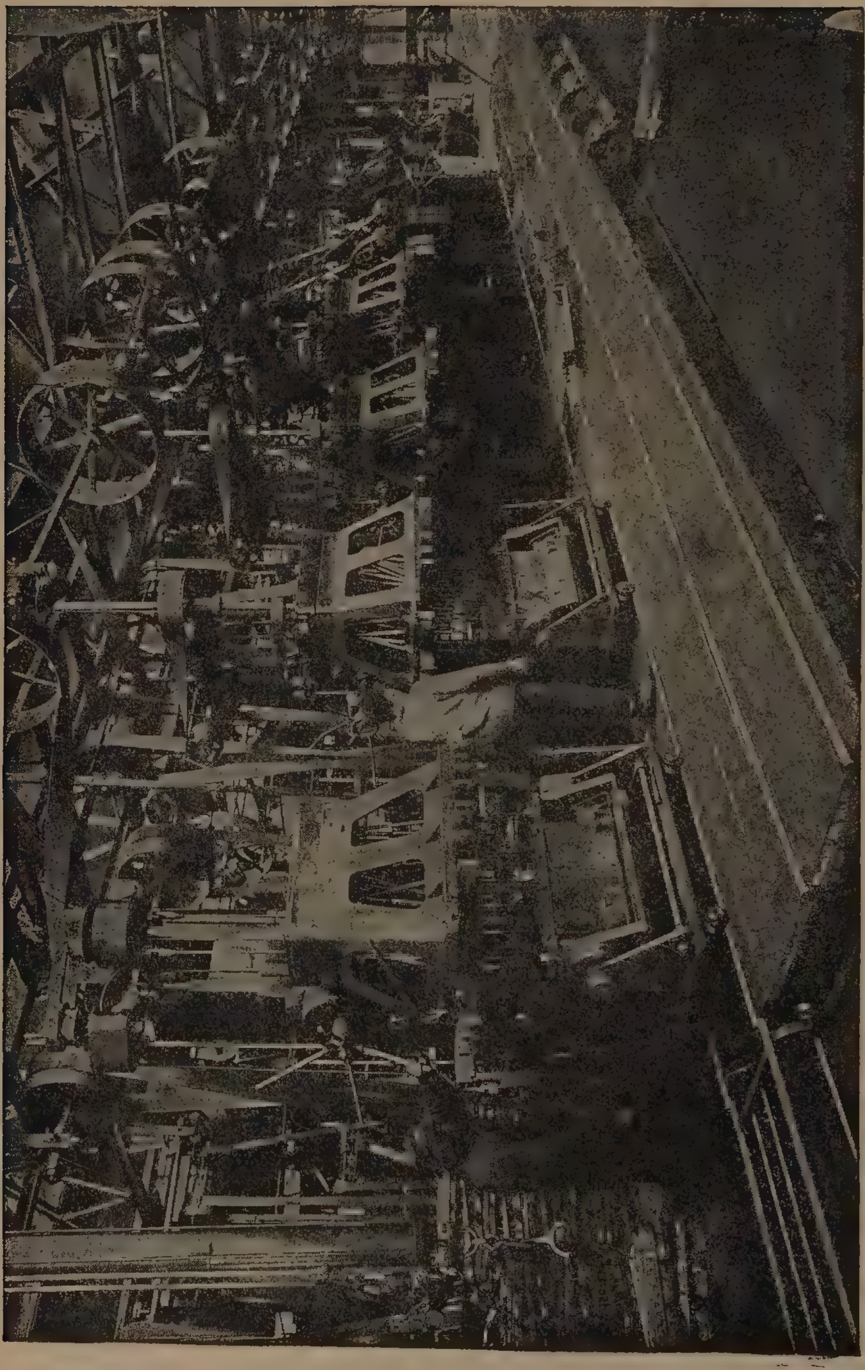


FIG. 76.—DRILLING, TAPPING AND COUNTER-BORING CYLINDER BLOCK IN RAILROAD GANG DRILL.

Fig. 76. This is a view from the plant of the Continental Motors Corp. A battery of multiple spindle drills are arranged in line and with a double steel rail track in the floor in front of them. Each cylinder block, being clamped to a fixture supported on a wheel truck, is operated on successively in each of the drill presses. The advantage of this lies in the fact that quite a number of drilling and similar operations have to be performed on the valve seats and adjacent parts, all on the same axis (there were still more of them when integral heads with valve caps were used), so that they could not be performed simultaneously. Thus the hole in or for the valve stem guide must be drilled and reamed, the valve port bored out, the valve seat faced, etc. In the Continental plant each cylinder casting is clamped into the fixture shown, known as a tumbling jig, and after eight operations it arrives at the end of the first track, completely drilled and tapped. The fixture is then switched over to the second or return track by means of the truck seen in the foreground, and a new cylinder block is inserted while it is being returned to the beginning of the line.

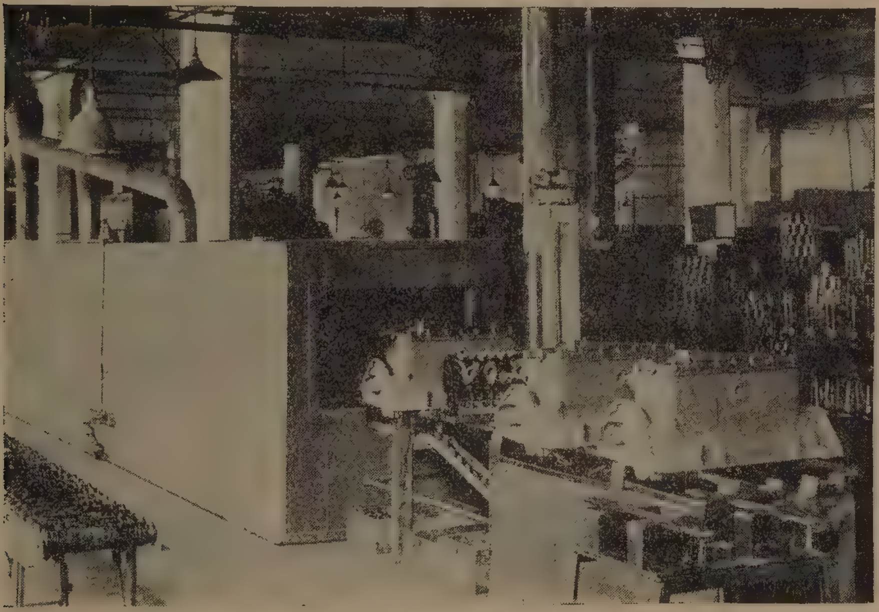


Fig. 77.—Cylinder Blocks Entering Enameling Oven.

Valve seats are sometimes hand-reamed and the valves are then ground in in a valve grinding machine. The fit or tightness of the valves may be determined by means of an air pressure test, for which purpose a head must be put on the cylinder block.

Machining Compression Space—To insure accuracy of compression spaces, the latter must be completely machined. One manufacturer of an L-head engine mills out that portion of the compression chamber in the head directly over the cylinder bore by means of a milling cutter of the full diameter of the chamber, guiding it by a substantial bushing. He then profiles the head for valve clearance and also profiles all around the combustion chamber, using the clamping plate as a guide for the profiling tool. The guiding surfaces on the plate are made of steel and in sections, and a roller on the spindle of the cutter contacts with them. Profiling is also done on the edges of the valve chamber.

After the cylinder blocks and heads are completely machined they are washed in a cleaning solution to free them of all dirt and grease, and they are then sprayed with japan and baked, rubber caps being placed over all parts that are not to receive a coating of paint, while the block is being airbrushed. One manufacturer has recently begun to give the blocks a cellulose lacquer finish which is harder and more resistant to scratching and does not require oven-baking.

CHAPTER VII

Piston, Piston Rings and Piston Pin

The piston in a gasoline engine serves a triple function. It forms the movable wall of the combustion chamber, allowing that chamber to increase and decrease in volume; it receives the pressure of the expanding gases and transmits this pressure to the connecting rod, and it forms a crosshead through which the side thrust due to the angularity of the connecting rod is transmitted to the cylinder wall.

The piston makes of the combustion chamber a closed vessel of variable capacity. Since the pressure within the combustion chamber is sometimes as high as 400 lbs. per sq. in., a high degree of gas-tightness is essential. The cylinder, as already stated, is bored to form a true cylinder as nearly as possible. If the piston were turned to exactly the same diameter as the bore, over its whole length, it might form a fairly gas-tight joint; but this is impractical, owing to differences in heat expansion. The piston head heats more than the cylinder wall, because it is not water-jacketed, and the head end of the piston also heats more than the open end, which latter is not exposed to the heat of the burning gases and receives heat only by transmission. These facts must be taken into consideration in designing the piston, which must be made of less diameter at its head end than at its open end, and of less diameter than the bore at its open end. If it were made of the same diameter as the bore it would be subject to undue friction even when cold, and it would seize when the engine heated up. Since the piston must be made smaller in diameter than the cylinder bore it cannot be depended upon to ensure gas-tightness; this quality is attainable only by means of flexible, split metallic rings, called piston-rings, three or four of which are placed upon the piston in grooves turned for the purpose in its outer circumference.

Piston Temperatures—Explorations of piston temperature under working conditions, by means of thermo-couples, have been made by Frank Jardine and Ferdinand Jehle, on aluminum pistons in a single cylinder Liberty engine (5 in. bore), and by Dr. Riehm on a Diesel engine cast iron piston of 14 in. diameter. The results of both series of tests show that the hottest point is the center of the piston head, that the temperature decreases slowly toward the edge of the piston head and rapidly down the side of the piston over what may be called the ring belt. In the Jardine-Jehle tests thermo-couples were inserted at the center of the piston head, about two-thirds the way out to the edge, directly below the lowest ring groove (above the piston pin) and close to the lower edge of the piston skirt. The temperature distribution in the piston will, of course, depend upon the design of the piston and upon the working conditions. It would seem from the two sets of experiments that with the engine operating at nearly full load and the cooling water entering the jacket at about 150° Fahr., the temperature at the center of the piston head in an average Otto-cycle engine is around 700°; at the circumference of the piston head, about 550°; in the skirt just below the ring belt, 275°, and at the open end of the piston, 200°. It will thus be seen that there is a great drop in temperature from the top to the bottom of the ring belt, and as most of this space is occupied by the rings, which are in intimate contact with the cylinder wall all around, most of the heat, no doubt, passes to the cylinder wall through them. From the fact that the piston rings are in contact with the cylinder wall at about 200° and with a portion of the piston at from 275° to 550°, one might expect them to attain a temperature of about 400° under full load operating conditions. Practice, however, has shown that a gap clearance of 0.0025 inch per inch of bore is sufficient to prevent injury to the ring from "butting," and since cast iron expands 0.000556 inch per inch per 100° Fahr., the temperature difference cannot exceed

$$\frac{0.0025 \times 100}{0.000556 \times 3.14} = 150°.$$

Some makers allow a gap clearance of 0.003 inch per inch bore, while others specify for the top ring a slightly greater clearance than the first figure given.

Forms of Gap—There are two forms of piston ring gap in common use, the diagonal or scarf gap and the lap joint gap, both of which are illustrated in Fig. 78. In addition to these there are many special forms of gap designed to prevent leakage, but the two mentioned are in most extensive use. It is directly apparent that with a diagonal gap there is a leakage path for the gas if the ends of the ring do not come close together. Some slight space must be left between the ends of the ring, and it was pointed out in the foregoing that a circumferential gap clearance of 0.0025 in. per inch of bore had been found sufficient. With a 45° diagonal gap, the width straight across the gap corresponding to this circumferential gap clearance is

$$0.0025 \times \sin 45° = 0.0025 \times 0.707 = 0.00175 \text{ in.}$$

per inch of bore, and with a 30° diagonal gap it is 0.00125 inch per inch of bore.

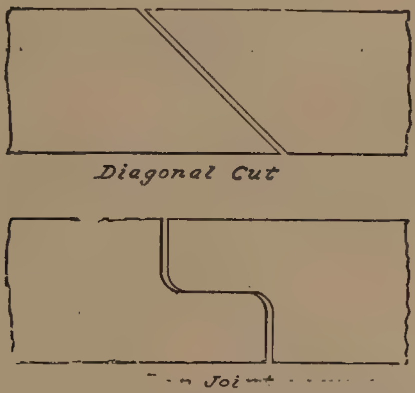


FIG. 78.—PISTON RING JOINTS.

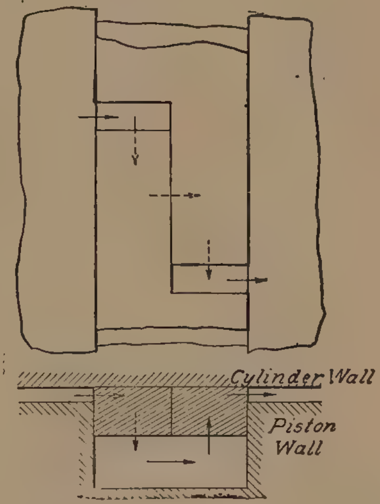


FIG. 79 (RIGHT).—LEAKAGE PATH AROUND LAP-JOINTED PISTON RING.

With a lap-jointed ring there is also a small leakage path for the gas at the joint. This is illustrated in Fig. 79. The piston must be given a considerable clearance at the top end, or over the ring belt, and this clearance is shown exaggerated in the illustration. The tongues or overlapping portions of the ring also do not quite fill up the groove, especially if the ring is of the eccentric type, in which case the depth at the ends of the ring is only about one-half the depth of the groove. The gas, therefore, passes from the piston clearance space into the gap clearance space, down the latter to the bottom of the ring groove, thence into and up the other gap clearance space and into the piston clearance, as indicated by the arrows.

Engineers in general seem to be of the opinion that the leakage at the joint is a small item in the case of any well-fitted ring. When leakage does take place it is generally either between the ring and the cylinder wall, owing to an improper fit of the ring at certain parts of its circumference, or through the ring groove, owing to excessive side clearance. The side clearance of the ring in the groove increases in the course of time, due to wear, especially in the case of pistons made of soft material such as aluminum alloy, and the clearance therefore should be made as small as practical in the first place. It is customary to allow a side clearance of 0.001 in., but some makers, by selective assembling, limit the clearance to between 0.00025 and 0.00075 in. The position of the ring in the groove is determined by the friction between it and the cylinder wall and by the inertia of the ring, and it is evident that the ring shifts from one side of the groove to the other at least once per stroke.

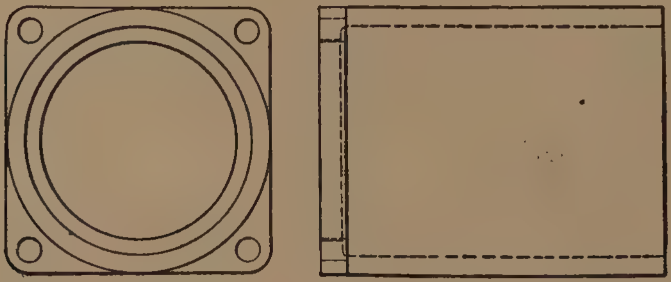


FIG. 80.—POT FROM WHICH PISTON RINGS ARE CUT.

Material of Rings—There are two qualities which a material for piston rings must possess. It must be tolerably elastic, so that the ring may exert the necessary pressure against the cylinder wall, and may be expanded to be stripped over the piston while being inserted in the groove, and it must be comparatively soft, so it will not score or unduly wear the cylinder wall. Cast iron has been found to be the most suitable material, but the best grade of close-grained gray iron should be used. Formerly piston ring blanks were often cast from the same "run" in the foundry as cylinders and pistons, but the manufacture of piston rings has now been largely specialized, and it has been found that an iron with somewhat less silicon and more sulphur than the ordinary cylinder iron gives better results.

Individual vs. Pot Castings—The original method of making ring blanks consisted in making "pot" castings, that is, long cylinders, from which, after turning and boring, the rings were cut off in the lathe (Fig. 80). At the present time, however, a large proportion of all rings are "individually" cast. That is to say, the pattern for the ring is similar in shape to an individual ring, but a number of such patterns are "gated," and several gates are stacked one above the other, so that a considerable number of ring blanks are produced in a single pouring. The pattern for each ring is formed with a "run-over" and the corresponding portion of the mold fills up with metal before the ring portion proper is filled up. Hence, any loose sand, etc., is carried into this "run-over" and the ring itself consists only of sound metal. A "gate" of "individually" cast rings is shown in Fig. 81.

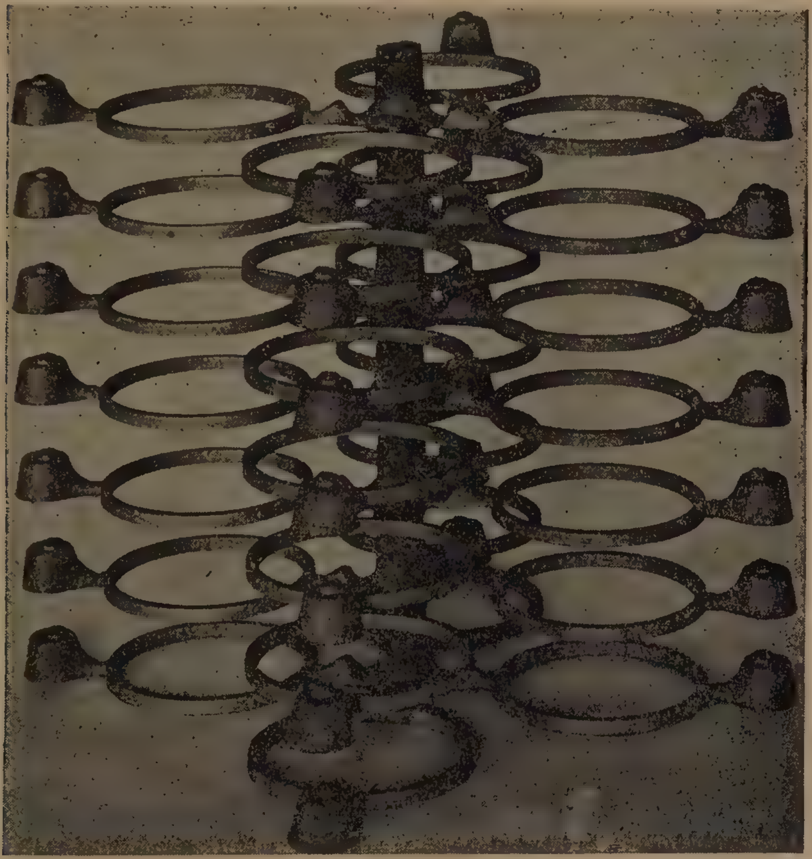


Fig. 81.—Gate of Individually Cast Ring Blanks.

Rings that are individually cast are left in the rough on the inside, and the chief advantage claimed for them is that the metal immediately below the scale is of a denser and more "springy" nature than the metal further inside the casting. To increase the density of the cast iron and thus improve its elasticity, it has even been attempted to cast rings in molds that are rotating at high speed and thus subject the iron while filling the mold and cooling, to centrifugal force. Centrifugally cast rings are made in England.

Stresses in Piston Rings—In Fig. 82 is represented a section A' B' C' D' of a piston ring of an outside radius r_1 when in the free state, and a thickness t. When the ring is inserted into the cylinder it is compressed to a radius r, which is, of course, the radius of the cylinder bore. The actual change in the radius is comparatively slight, and some of the dimensions in the figure are exaggerated in order to more clearly bring out the effects of compressing the ring. The fibers along the outer circumference of the ring will be extended and those along the inner circumference compressed, the section assuming the form A B C D. It is shown in textbooks on the mechanics of materials that the relative extension of material under tension and its compression under pressure are equal to the stress S to which it is subjected divided by its coefficient of elasticity E. Therefore, referring to Fig. 82, let S be the compressive stress on the inner fibers and $-S$ the tensile stress on the outer fibers of the ring. Since the ring is subjected to plain bending stresses the compression and tension are equal, and we may write

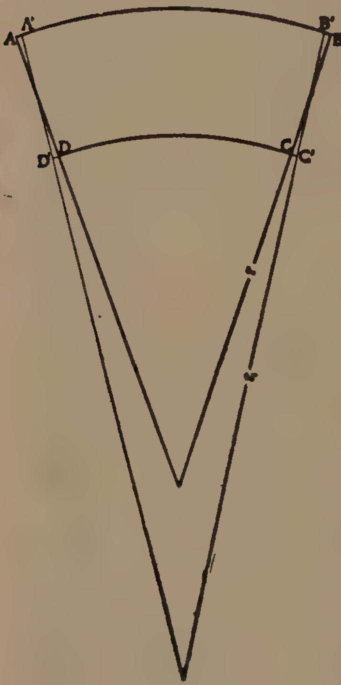


Fig. 82.

$$\frac{-S}{E} = \frac{A'B' - AB}{AB} = \frac{A'B'}{AB} - 1,$$

and

$$\frac{S}{E} = \frac{D'C' - DC}{DC} = \frac{D'C'}{DC} - 1.$$

Transposing,

$$\frac{A'B'}{AB} = \frac{-S}{E} + 1. \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

$$\frac{D'C'}{DC} = \frac{S}{E} + 1. \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

But

$$AB : DC = r : r - t,$$

and

$$A'B' : D'C' = r_1 : r_1 - t.$$

Dividing the lower proportion by the upper

$$\frac{A'B'}{AB} : \frac{D'C'}{DC} = \frac{r_1}{r} : \frac{r_1 - t}{r - t}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

Substituting the values of $\frac{A'B'}{AB}$ and $\frac{D'C'}{DC}$ found in (39) and (40)

$$\frac{-S}{E} + 1 : \frac{S}{E} + 1 = \frac{r_1}{r} : \frac{r_1 - t}{r - t};$$

$$\left(\frac{-S}{E} + 1\right)\frac{r_1 - t}{r - t} = \left(\frac{S}{E} + 1\right)\frac{r_1}{r};$$

$$\frac{r_1 - t}{r - t} - \frac{r_1}{r} = \frac{S}{E}\left(\frac{r_1}{r} + \frac{r_1 - t}{r - t}\right);$$

$$\frac{S}{E} = \frac{\frac{r_1 - t}{r - t} - \frac{r_1}{r}}{\frac{r_1}{r} + \frac{r_1 - t}{r - t}}$$

$$= \frac{r(r_1 - t) - r_1(r - t)}{r_1(r - t) + r(r_1 - t)}$$

$$= \frac{t(r_1 - r)}{2rr_1 - t(r + r_1)}.$$

Consequently

$$S = E\frac{t(r_1 - r)}{2rr_1 - t(r + r_1)}. \quad . \quad . \quad . \quad . \quad . \quad (42)$$

This equation gives the stress in the outermost fibers of the ring when it is introduced into the cylinder. If the ring is to be subjected to the same stress when stripped over the piston as when compressed in the cylinder, its radius r_1 when in the free state should be a mean between the radius r when in the cylinder and the radius r_2 when expanded in stripping it over the piston. If the ring is a true annulus, that is, if its bounding lines are true circles, when in the cylinder, then its inner and outer circumferences are not true circles either when the ring is in the free state or when it is expanded while being stripped over the piston. In speaking of the radius r_1 of the ring in the free state and the radius r_2 of the expanded ring, what is meant is the average radius. If the ring could be expanded so as to remain a perfect annulus, then the outside radius r_2 to which it would have to be expanded in order to pass over the piston is equal to $r + t$. In reality the ring, when expanded by applying opposing tangential forces to its ends, as shown in Fig. 83, will assume an elliptical outline, owing to the fact that the bending moment is a maximum opposite the gap, and it is therefore at this point that the ring bends the most. This tendency to localization of the bending can be counteracted by applying pressure to the ring radially inward about 75° from both sides of the gap. By resorting to this expedient, the ring need not be expanded to an appreciably greater radius than $r + t$ in order to get it into the groove. We then have

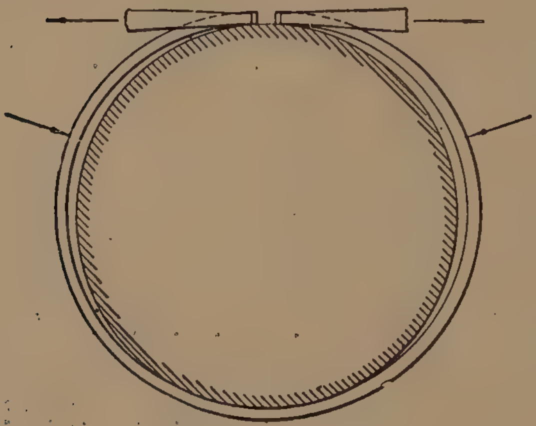


FIG. 83.—STRIPPING RING OVER PISTON.

$$r_2 = r + t,$$

and

$$r_1 = r + \frac{t}{2}.$$

Using this latter relation in order to eliminate r_1 from equation (42) we get

$$S = E\frac{\frac{t^2}{2}}{2r^2 - (rt + t^2/2)}$$

$$= E\frac{t^2}{4r^2 - (2rt + t^2)}.$$

Since $r = b/2$, where b is the cylinder bore, the above may be written

$$S = E\frac{t^2}{b^2 - (bt + t^2)}.$$

In view of the fact that in actual piston rings t is only about one-thirty-second of b, the expression $-(bt + t^2)$ in the denominator of the right-hand member of the equation can be neglected without causing any serious error, and the equation then becomes

$$S = E\frac{t^2}{b^2},$$

which may be transformed to read,

$$t = b\sqrt{\frac{S}{E}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

Pressure Against Cylinder Wall—This pressure is radially outward and should preferably be uniform over the entire surface of the ring. It is balanced by an equal and opposite reaction of the cylinder wall against the ring. We will denote this reaction per square inch by p, and the width of the ring by w.

The reaction of the cylinder wall on the ring produces a bending effect in the latter which subjects the material of the ring to a certain stress. The pressure of the ring against the cylinder wall and the stress in the material of the ring are mutually related, and we will now investigate this relation. We will take a cross section of the ring at C, Fig. 84, and find the bending moment for that section.

Consider a small strip da of the ring surface between A and C. The area of this strip is evidently $w\ da$, and the total pressure on it, $p\ w\ da$. This pressure can be divided into two components, one, $p\ w\ da \sin\theta$, parallel to the cord $A\ C$, and the other, $p\ w\ da \cos\theta$, perpendicular to the cord $A\ C$. It is directly apparent that, considering the pressure acting on the entire arc $A\ C$, the components parallel to the cord $A\ C$ vanish, and it is only the components perpendicular to $A\ C$ that produce a bending moment at C.

Considering the expression for this perpendicular component, we find that

$$da \cos\theta = dc,$$

because

$$\text{Angle } G\ E\ F\ (\theta) = \text{angle } H\ E\ M,$$

$H\ E$ being perpendicular to $E\ F$ and $E\ M$ perpendicular to $E\ G$, and

$$\text{Angle } H\ E\ M = \text{angle } E\ M\ C,$$

$H\ E$ being parallel to $M\ C$. The line $E\ M$ is a tangent to the arc $A\ C$ at E, and comprises, therefore, the short section da of this arc.

The component of the radial pressure on da perpendicular to $A\ C$ is therefore $p\ w\ dc$. That is, that component of the radial pressure on any small strip da across the circumference

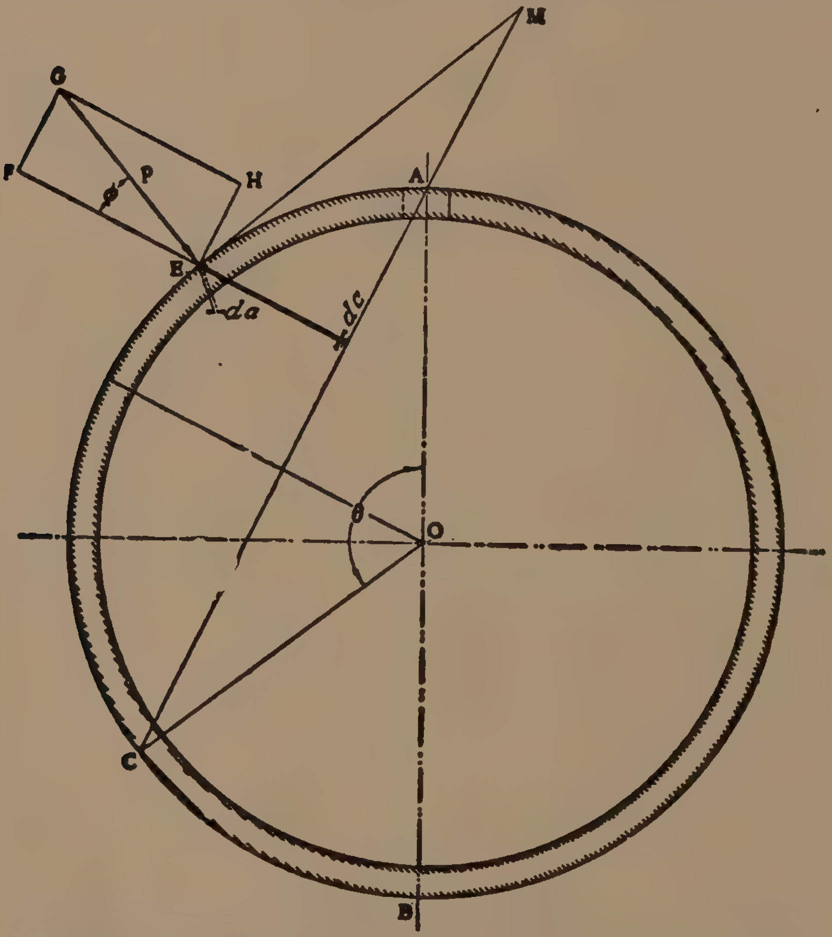


Fig. 84.—Diagram of Forces and Moments on Piston Ring.

of the ring which produces a bending moment at C is equal to the product of the area of the projection of that strip on the cord $A\ C$, into the radial pressure. As this applies to every small strip of the arc $A\ C$, it follows that the total pressure producing a bending moment at C is

$$\Sigma pwdc = pwAC.$$

But

$$AC = 2AO \sin \frac{\theta}{2} = 2r \sin \frac{\theta}{2}.$$

The above force acts on the point C through lever arms varying in length from zero to $2r \sin \theta/2$, the mean length being

$$l = r \sin \frac{\theta}{2},$$

consequently the bending moment at C is

$$M = 2pwr^2 \sin^2 \frac{\theta}{2} = \frac{pwb^2}{2} \sin^2 \frac{\theta}{2}.$$

The resistance to bending of a rectangular section of base w and height t is

$$M_r = \frac{SI}{c} = \frac{Swt^2}{6}.$$

The bending moment is equal to the resisting moment, and we may therefore write

$$\frac{pwb^2}{2} \sin^2 \frac{\theta}{2} = \frac{Swt^2}{6},$$

from which it follows that the radial pressure of the ring is

$$p = \frac{S}{3 \sin^2 \frac{\theta}{2}} \frac{t^2}{b^2}. \qquad \text{(44)}$$

From the above discussion it will be apparent that the bending moment on the ring under uniformly distributed pressure increases from nothing at the gap to the maximum directly opposite the gap, and that if the section is uniform, in other words, if the ring is of the concentric type, the stress in the metal will vary from nothing at the gap to the maximum at the point opposite the gap. From the standpoint of the maximum pressure which a ring can exert against the cylinder wall it is therefore this latter part of the ring which deserves special attention. For the point opposite the gap, θ is equal to 180°, $\theta/2$ is 90° and $\sin \theta/2$ is 1. Hence, for that point

$$p = \frac{St^2}{3b^2}.$$

We found in the foregoing that

$$\frac{t^2}{b^2} = \frac{S}{E},$$

and substituting in the above we get,

$$p = \frac{S^2}{3E}. \qquad \qquad (45)$$

The coefficient of elasticity E for cast iron is about 14,500,000. The tensile strength of good gray cast iron is about 34,000 lbs. per sq. in., and that of "individual" castings is probably somewhat greater. A working stress S of 18,000 lbs. per sq. in. can be allowed, and this gives for the specific pressure obtainable,

$$p = \frac{18{,}000 \times 18{,}000}{3 \times 14{,}500{,}000} = 7.5 \text{ lbs. per sq. in.}$$

In the foregoing it was shown that

$$t = b\sqrt{\frac{S}{E}},$$

and with $S = 18{,}000$ lbs. per sq. in. and $E = 14{,}500{,}000$ we get

$$t = b/28.4$$

We assumed that when the ring is in the free state its average radius is $r - t/2$; hence, the combined length of the ring and gap is $2\,\pi\,(r - t/2)$, and since the length of the ring in the compressed state is $2\,\pi\,r$, the length of the gap is

$$2\,\pi \times t/2 = \pi\,t$$

Substituting the value of t found above we find for the length of the gap,

$$l = b/9,$$

to which must be added the gap clearance $0.0025\,b$, making the total length of the gap when the ring is in the free state, $0.1136\,b$.

Methods of Putting Ring under Tension—There are two fundamentally different methods of putting rings under tension or causing them to exert pressure against the cylinder wall. One consists in originally turning the ring to a slightly larger outer diameter than the cylinder bore, cutting out a section of the ring and then, after bringing the ends nearly together, turning the thus compressed ring to the diameter of the cylinder bore. Such a ring should fit the cylinder bore, but theory indicates, and experience has proven, that it bears against the cylinder wall very unevenly, the greatest pressure being exerted at the ends and in the middle. In order to equalize the pressure to a certain extent, the rings are sometimes made eccentric, the thickness at the gap being only one-half that opposite the gap. This only partly solves the problem of uniform pressure, which would theoretically require reducing the thickness to nothing at the ends of the ring, and, besides, owing to the small depth of the ring near the gap, the danger of leakage between the ring and the side of the groove is increased, particularly because the side of the groove is worn more rapidly by the narrow ends of the ring.

The other method consists in turning the ring to the diameter of the cylinder bore in the first place and then placing it under tension by some mechanical means. The most common method consists in hammering or peening the ring on the inside after it has been "gapped." Peening of rings was introduced by Capitaine, one of the pioneers of the gas engine industry, and has been greatly developed in connection with rings for automotive engines. The peening spreads the metal on the inside of the ring, and, in consequence, increases the radius of the ring, opening up the gap.

Instead of by peening, rings are sometimes opened or put under tension by rolling or knurling them under pressure on the inside, while in a die or mold of suitable form, or the ring, after being gapped, may be expanded over a mandrel of suitable size and shape, and while in the expanded form, subjected to a heat-treatment, which causes the molecules of the iron to set in the positions they occupy while the ring is in the expanded condition, so that it remains open at the gap and exerts pressure against the cylinder wall when introduced into the cylinder.

Expanded Rings—The third method consists in either casting or turning the ring to what is known as the expanded form. This is such a form that when a uniform radial pressure is applied to the outside of the ring after the gap has been cut, it will be compressed to a perfect circle of the diameter of the cylinder bore. This form can be calculated, at least approximately. The form is non-circular and can therefore be turned only with the aid of a master ring or cam. A non-circular curve is composed of a very large number of small circular arcs which are defined by their radius, the latter being referred to as the radius of curvature at that part of the curve. Owing to the fact that at the ends of the ring the bending moment is zero, the radius of curvature there will not be changed by the reaction of the cylinder wall against the ring, and the radius there should be made equal to that of the cylinder bore. As we go around the ring from one end, the bending moment increases and the primitive radius r_1 of the ring therefore should increase, until it attains a maximum value directly opposite the gap. Just how it should increase and what its maximum value should be can be determined by the following considerations.

Form of Expanded Ring—By a transformation of equation (44) we get an expression for the stress S at any section of the ring—

$$S = 3p\frac{b^2}{t^2}\sin^2\frac{\theta}{2}.$$

We have another expression for this stress in terms of the primitive radius r_1, the "compressed" radius r and the thickness t in equation (42)

$$S = E\frac{t(r_1 - r)}{2rr_1 - t(r + r_1)}.$$

Equating the two expressions for the stress we get,

$$3p\frac{b^2}{t^2}\sin^2\frac{\theta}{2} = E\frac{t(r_1 - r)}{2rr_1 - t(r + r_1)}.$$

Dividing both sides of the equation by t, multiplying by $2r$ $(= b)$ and at the same time substituting $2r$ for b, we get

$$24p\frac{r^3}{t^3}\sin^2\frac{\theta}{2} = E\frac{r_1 - r}{r_1 - t/2 + (t/2)(r_1/r)}.$$

In order to be able to derive an expression for the primitive radius r_1, it is necessary to simplify the right-hand side of this equation, and this can readily be done without introducing an appreciable error. In all practical rings t is very close to $r/16$ and r_1 is very nearly the same as r. Inserting these equivalents in the above equation we get,

$$24p\frac{r^3}{t^3}\sin^2\frac{\theta}{2} = E\frac{r_1 - r}{r_1 - r_1/32 - r_1/32}$$

$$= E\frac{r_1 - r}{\frac{15}{16}r_1}$$

$$= \frac{16}{15}E\frac{r_1 - r}{r_1}.$$

Therefore,

$$22.5p\frac{r^3}{t^3}\sin^2\frac{\theta}{2} = E - E\frac{r}{r_1},$$

$$E\frac{r}{r_1} = E - 22.5p\frac{r^3}{t^3}\sin^2\frac{\theta}{2},$$

$$Er = Er_1 - 22.5r_1p\frac{r^3}{t^3}\sin^2\frac{\theta}{2},$$

$$Et^3r = Et^3r_1 - 22.5r_1pr^3\sin^2\frac{\theta}{2},$$

$$r_1 = \frac{Et^3r}{Et^3 - 22.5pr^3\sin^2\frac{\theta}{2}}. \qquad (46)$$

The above equation defines the form of the outside circumference of an expanded ring pattern, except for the finish allowance.

To those who are not familiar with mathematical methods of reasoning it may be of help to state that when a ring is expanded by hammering, etc., or is made to an expanded pattern, the radius of curvature of the outside surface of the ring when in the free state, opposite the gap should be substantially equal to the radius of the cylinder bore plus the ring thickness, and the radius of curvature should decrease gradually from this point toward both ends of the ring, being equal to the radius of the cylinder bore at the very ends. (Fig. 85)

In hammering a ring the greatest change in the curvature must be made at the point opposite the gap, and no change at the ends. Hence the intensity of the hammering must increase gradually from the ends of the ring towards the middle. In manufacturing hammered rings the hammer blows are regulated automatically, and these rings are being made only by specialists.

Quick-Seating Rings—Originally piston rings for automobile engines were ground to a smooth surface on the outside, but it has been found that they wear down to a good seat in the cylinder bore much quicker if they are turned on the outside, and it is now common practice to turn them, some makers even taking a rather coarse cut. Another plan to ensure quick seating consists in under-cutting one half or two thirds of the outer surface to a depth of about 0.002 in.

Width of Rings—It is desirable to make the rings narrow, because then the loss of power due to friction between them and the cylinder wall will be less, and there probably will also be less wear on the sides of the ring grooves. On the other hand, narrow rings have the disadvantage that they are somewhat more delicate to handle. It has been customary to make the width of the ring about one-twentieth of the bore, and rings 3/16 in. wide are largely used for automobile engines of between 3 and 4 in. bore, but a number of prominent makers use ⅛ in. rings for bores within this range.

Laterally Expanding Rings—In old engines leakage frequently occurs between the ring and the sides of the ring groove, and a number of piston rings are now made that expand laterally as well as radially and thus take up any excess clearance there may be as a result of poor fitting or wear. These rings consist of either two or three parts. Where three parts are used one is generally a steel ring which supplies the pressure, but does not come in contact with the cylinder wall. An incidental advantage claimed for these rings is that they tend to prevent piston slap. Sections of several such rings are shown in Fig. 86.

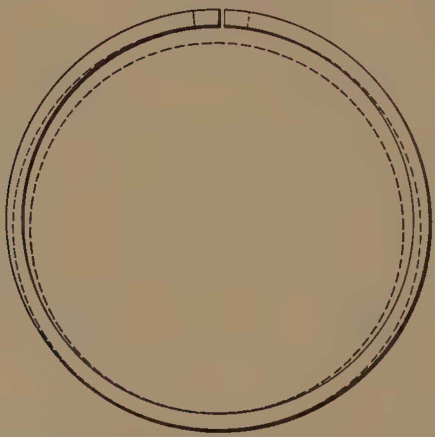


Fig. 85.—Relation of Expanded Ring to Ring Groove.

Special Piston Rings—Considerable ingenuity has been exercised in recent years in the elimination of the gap in the ring, which is one of the causes of leakage. Abroad a ring comprising two complete turns, known as the Lehmann double spire ring (Fig. 87), seems to have considerable vogue. It is a concentric ring and apparently is cut with a very fine circular saw after the blank has been turned up and cut off. In this country the leakproof piston ring, shown in Fig. 88, has been on the market for some years. It consists of two separate rings, one having an outward and the other an inward flange, the flange on one ring obturating the slot in the other. Both rings are turned concentrically, and uniform pressure is aimed at by placing the slots on opposite sides, so that the non-uniformity of the outward pressure of one ring is compensated by that of the other. The separate rings are prevented from sliding upon one another by a pin, but the complete ring is free to turn in the groove.

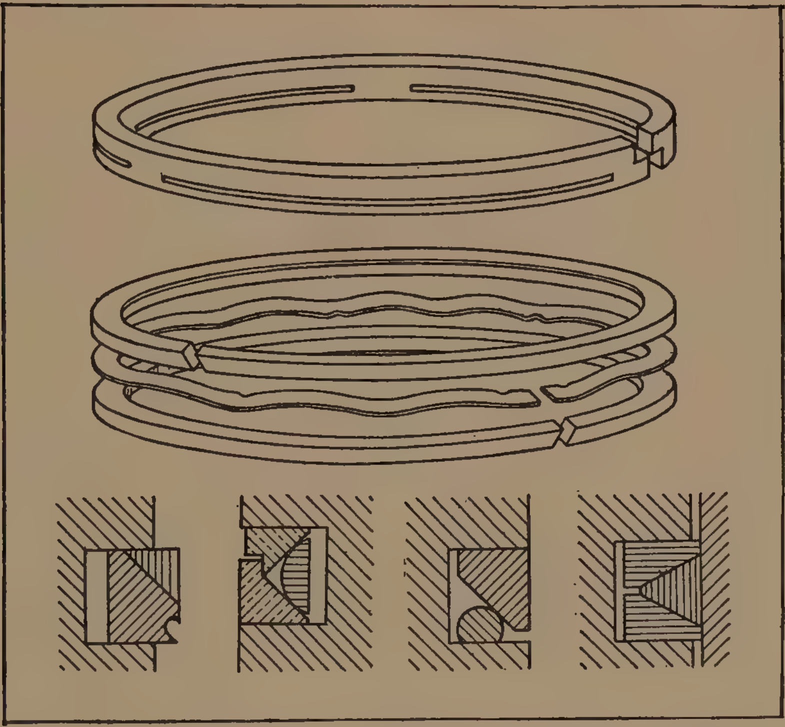
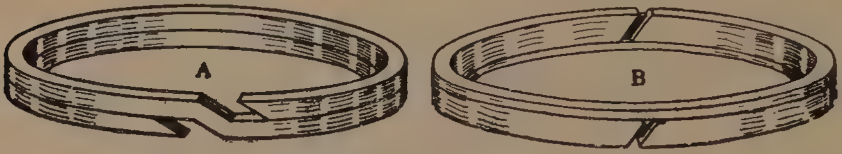


Fig. 86.—Examples of Laterally Expanding Rings.

Oil Rings—Each piston of an engine is now generally fitted with one oil ring which scrapes excess oil off the cylinder wall and returns it to the crankcase through holes drilled in the piston skirt. This ring is placed in the first ring groove above the piston bosses. The characteristic which usually distinguishes the oil ring from the other rings is that its lower edge is quite sharp, and there may be a channel formed on the ring which receives the oil that is scraped off the cylinder wall. Fig. 89 shows a form of oil "regulating" ring that has come into wide use in recent years. It is turned with a central oil groove all around its circumference, and a number of slots are milled entirely through the ring, through which the oil can pass into the bottom of the ring groove, from which latter it passes through holes drilled in the piston wall back to the crankcase. One probable reason why this type of oil control ring has largely replaced the type with a sharp edge which scrapes the excess oil into a groove cut in the piston skirt adjacent to the groove for the oil ring is that with the latter construction the bearing surface at one side of this ring is greatly reduced, hence the side clearance of the ring increases more rapidly.



Figs. 87 and 88.—Special Forms of Piston Ring.

Grinding Rings—It is now common to grind piston rings on both sides, and in cutting off the rings the proper allowance must be made for this grinding. The allowance in width for grinding on the sides should be from twenty to thirty one-thousandths of an inch. Grinding of the sides of the ring is done on a special piston ring-grinding machine on which the rings are held by means of a magnetic chuck. Such a grinder is illustrated in Fig. 90.

Next the rings are "gapped." This operation differs, of course, according to whether the ring is to have a diagonal joint or a lap joint. In either case the cutting is done on a hand miller with a cutter of exactly the right width for the slot to be cut. Before doing any further work upon the rings, it is best to submit them to a test, so as to minimize the waste due to rejections. The easiest way to make this test consists in forcing the ring over a tapered

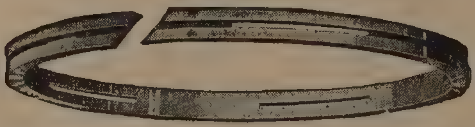


Fig. 89.—"Perfect Circle" Oil Control Ring.

arbor to a certain mark and rejecting all rings which show more than a certain permanent set, determined by measuring the width of the gap before and after the test.

Next the rings are turned or ground on the outside, and in order that they may fit the cylinder after this operation they are first compressed in a fixture, then clamped on an arbor while in the compressed state, and finally ground down to the exact diameter of the cylinder bore. Fig. 91 shows a form of arbor on which the rings may be ground. This arbor is provided with two collars. The one on the left is of such a diameter that the clamping fixture will just pass over it, by which means the rings are centered on the arbor. The collar on the right-hand side is of the exact diameter of the cylinder bore to which size the rings are to be finished. The collar on the left may be integral with the arbor, but if it is loose it should be held from turning by a pin or key, as should the other collar. This collar is drawn up tightly by means of a hexagonal nut, the clamp

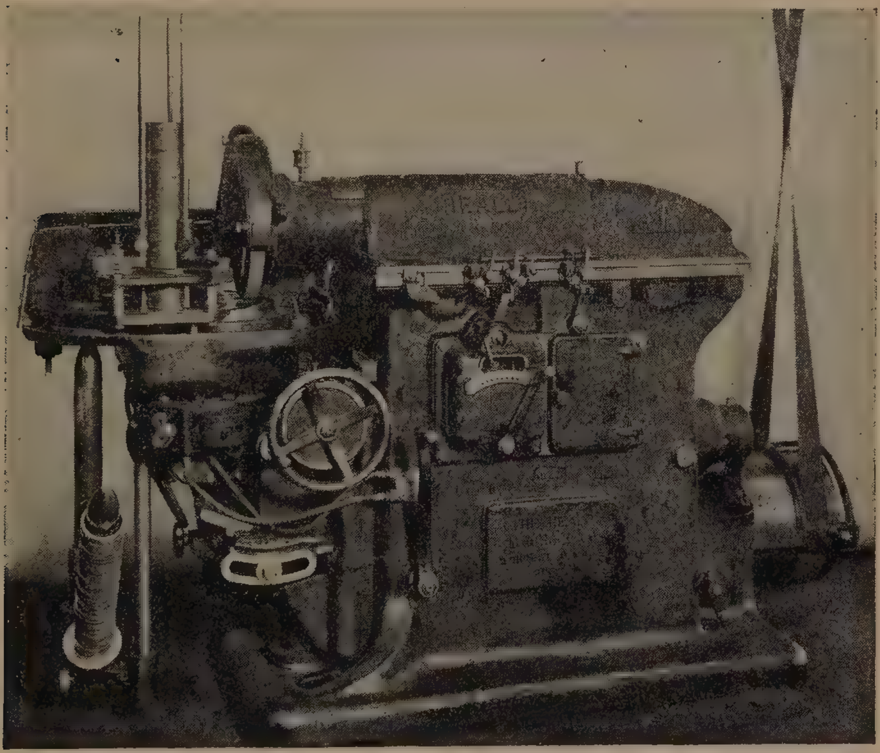


Fig. 90.—Heald Automatic Piston Ring Grinder.

is then removed, the arbor is placed in the lathe or grinding machine and the rings are ground to size.

Testing Piston Rings—A common method of testing piston rings is known as the light test. The ring is put into a ring gage of the diameter of the cylinder bore, the gage being placed over an opening in the front of a box containing an electric bulb. If no light shows through between the gage and the ring at any part of the circumference, the ring is considered good. This test proves the fit of the ring, but it is not conclusive as to its merits, as the ring may fit parts of the cylinder wall and still not exert any pressure upon them.

The total pressure exerted by the ring against the cylinder wall and the average pressure per square inch can be determined by means of a very simple apparatus (Fig. 92). One end of the ring is clamped, by means of a C clamp, to a horizontal arm projecting from a standard, and a thin cord or wire secured to another horizontal arm extending from a point higher up on the same standard, is wound once around the ring and has a small pail or scale pan tied to its lower end. Into this pail or pan shot or other weights are placed until the tension on the cord suffices to close up the ring to the same degree as when in the cylinder. For accurate results it is essential that the cord or wire be well oiled. Let W represent the total weight on the cord when the ring is compressed to the standard diameter, and let l be the length of the gap taken up. Then, since the tangential force applied to close up the ring increased from nothing to W, the work done in closing it up is $W\,l/2$. When the ring is closed up by the reaction of the cylinder wall, which we will assume to be equal to p pounds per square inch of contact surface, the decrease in the average radius of the ring is $l/2\pi$, and the work done, which is equal to that done in the test, is equal to

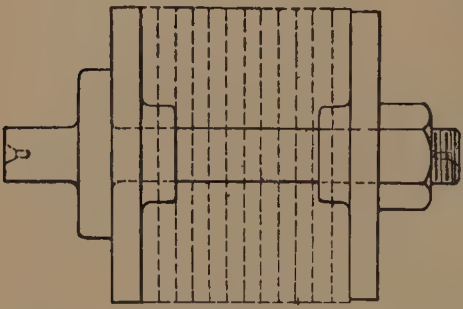


Fig. 91.—Mandrel for Turning Rings.

$$\frac{1}{2} \times \pi bwp \times \frac{l}{2\pi} = \frac{bwpl}{4}.$$

Equating this to the work done in the previous case (the test) we get,

$$\frac{bwpl}{4} = \frac{Wl}{2},$$

from which we get,

$$p = \frac{2W}{bw},$$

where p is the average pressure of the ring against the cylinder wall, in lbs. per sq. in., W, the weight required to close the ring, b, the cylinder bore and w the width of the ring, both in inches.

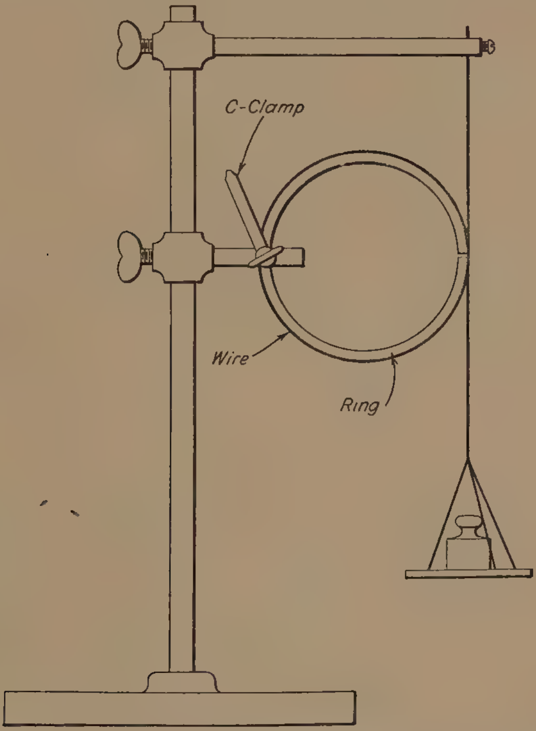


Fig. 92.—Simple Device for Determining Radial Pressure of Rings.

Saurer Piston Ring Tester—A device for rapidly comparing piston rings with a standard ring, or, in other words, for inspection service in the factory, has been developed by the firm of Adolph Saurer of Arbon, Switzerland. It compresses the ring to the diameter of the bore by exerting pressure at eleven equidistant points of the circumference, and then indicates whether the total pressure exerted by the ring is within certain permissible limits.

Referring to Fig. 93, mounted on ball-bearings inside the circular casing is a steel ring having radial slots in its upper face. A number of bell cranks are pivoted to the casing, and a roller on the outer end of each bell crank fits into one of the slots in the ring. The other end of each bell crank also carries a roller, which is caused to bear upon the circumference of the piston ring to be tested by the rotation of the steel ring already mentioned. The latter ring carries a projecting finger which engages with a plunger located in a sliding tube.

By pressing on the lever which projects from the casing, this tube is caused to move toward the gauge shown in the drawing, against the pressure of a helical spring. The plunger is pressed up against a shoulder in the same direction by a helical spring within the tube, and is therefore carried along with the tube until the resistance to its motion is sufficient to overcome the pressure of the spring acting upon it. This happens when the finger which is carried along by the plunger, has rotated the steel ring sufficiently to cause the rollers to exert a certain pressure on the piston ring.

The gauge of the instrument has two concentric dials, the reading of one being due to the motion of the tube and that of the other to the motion of the plunger. To set the instrument, a steel disc of the closed size of the ring to be tested is inserted, and the lever pressed until the rollers are in contact with it. The dial is then rotated until the pointer is opposite the zero mark. A piston ring of standard strength is then substituted for the disc, and the lever is again pressed. When the ring has been compressed to the standard diameter, the pointer will again point to zero, but before this occurs the inner dial will have been actuated by the differential motion of the plunger, and will take up a position depending upon the relation between the reaction of the piston ring and the initial compression of the spring which holds the plunger up to its work. This compression is adjustable, and when the machine is properly set, the effect of closing it on a ring of standard size and strength is to bring the point to zero on the size scale and between two limit marks on the strength scale.

Micrometer Test—Another check test for piston rings consists in closing them up to the standard diameter by means of a cord, or by inserting them in a standard ring gage, clamping the ends together, and withdrawing them, and then measuring the diameter across the gap and at right angles thereto by means of a micrometer. In automobile size rings the tolerance on the diameter thus measured is usually held to 0.010 in. The clearance of the ring in the groove is usually made 0.001 in., and the pressure of the ring can be measured and held within certain limits by means of the cord or wire method above described.

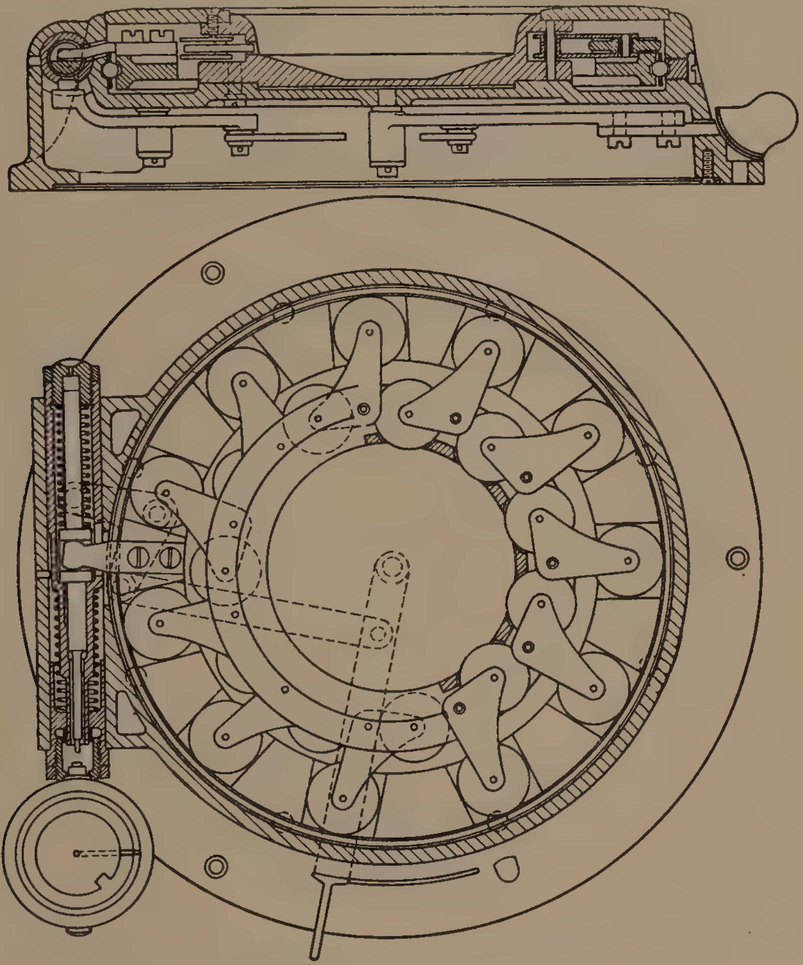


FIG. 93.—SAURER PISTON RING TESTING DEVICE.

Leakage Test—The effectiveness of the seal due to rings can be tested in service by means of a reproduction on a small scale of a gasometer as employed by gas works (Fig. 94). There are two tanks, one inside the other, with a water seal between them. The weight of the inner tank is balanced by a weight depending from a chain passing over a pair of pulleys. The crankcase is sealed and a connection is made from the breather to the gasometer. Any leakage of gas by the pistons passes from the crankcase through a pipe which enters the outer tank near the bottom and discharges into the movable one. A U-tube pressure gage on the latter shows that the pressure within it is maintained practically at atmospheric. As soon as the engine is started and a valve in the communicating passage opened, the inner tank begins to rise, and a scale on one of the standards by which it is guided, in connection with a stop watch, shows the rate at which leakage occurs.

The Piston Pin—The piston pin (or wrist pin, as it is often called) is the connecting link between the piston and the connecting rod. It is always made of tubular form and passes through bosses formed in the piston and through a hub at the upper end of the connecting rod. There are three general arrangements: The pin is either free to turn in both the piston bosses and the connecting rod, in which case it is said to float; it is secured against rotation and axial motion in one or both of the piston bosses and has a bearing in the connecting rod, or it is clamped in the top end of the connecting rod and has bearings in the piston bosses. The necessary dimensions of the piston pin are calculated by considering the pin first as a bearing journal carrying a certain load per square inch of projected area, and, secondly, as a beam supported at both ends and uniformly loaded between supports.

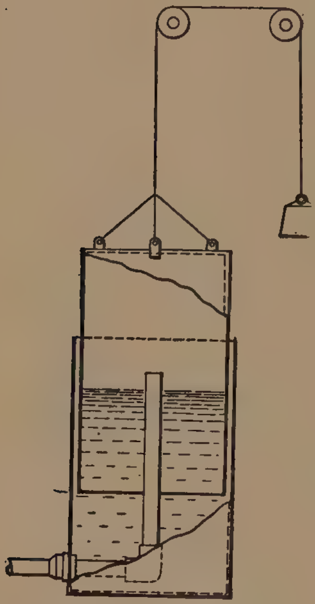


FIG. 94.—"GASOMETER" FOR MEASURING RING LEAKAGE.

Maximum Explosion Pressures—In determining the proper diameter of the piston pin (as well as other parts subjected to the force of explosion) account must be taken of the fact that different compression pressures are in use, and that in consequence the explosion pressures in different engines differ greatly. Kerosene-burning tractor engines operate at compression pressures of around 55 lbs. per sq. in. gauge, while some automobile engines operate at 80 lbs. gauge and over. The maximum explosion pressures in absolute measure are roughly proportional to the absolute compression pressures, and it is obvious that if the compression pressure is approximately 50 per cent greater in one engine than in another, the former must be made with stronger parts.

The maximum explosion pressure is a somewhat elusive figure, for it depends not only on the compression ratio but also on the form of the compression chamber, the location of the spark plug terminals, the intensity and timing of the spark, the temperature of the cylinder walls and possibly other factors. The maximum pressure undoubtedly occurs when, owing to defective ignition, several charges have been missed and the engine speed has been reduced, in consequence of which the compression space contains an unusually large charge of pure mixture. It is obvious that no very accurate figure can be given for the maximum explosion pressure, but the figures in the following table are fair approximations. The pressures given in the first column occur when the engine is running normally under full throttle at the speed of maximum torque, that is, at a comparatively low speed, while the figures in the second columns refer to the pressures developed if all of the dead gas is pumped out of the cylinder and the compression volume is then filled with a pure explosive mixture of air and gasoline vapor which is then ignited.

TABLE V

Compression Ratio (r)	Explosion Pressures (Lbs. per Sq. In.)	
	Normal	Maximum
3	200	300
3.2	220	320
3.4	240	340
3.6	260	360
3.8	280	380
4	300	400
4.2	320	420
4.4	340	440
4.6	360	460
4.8	380	480
5	400	500
6-1	500	

Calculation of Piston Pin—The length of the bearing portion of the piston pin should be as great as possible, but it is limited by the consideration that if the pin is fixed in the piston bosses the latter must be sufficiently long to provide room for a means for locking the pin, such as a set screw. In automobile engines the maximum length available for the bearing is equal to about one-half the bore. Therefore

$$l = \frac{b}{2}. \qquad \qquad (47)$$

With a maximum explosion pressure p per square inch, as determined from the above table, the total pressure on the piston is

$$P = \frac{\pi b^2 p}{4}.$$

When the engine is being started or is running at very low speed the pressure on the piston pin is the same as that on the piston. At high speed the former is considerably less than the latter, owing to the inertia of the piston, which opposes the force of the explosion. But the size of the piston pin should be governed by the maximum pressure sustained by it, and the inertia may therefore be neglected.

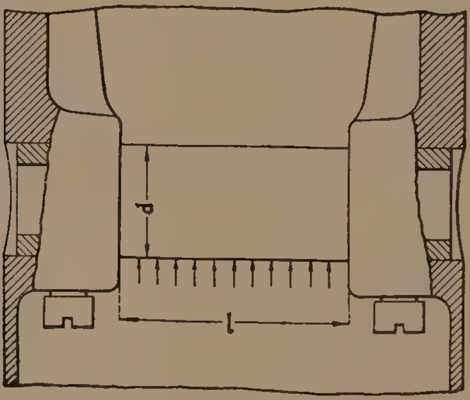


FIG. 95.—PISTON PIN LOAD DIAGRAM.

If we denote the diameter of the pin by d, then the projected bearing surface is

$$s = dl = \frac{db}{2},$$

and the maximum unit bearing pressure is

$$\frac{\frac{\pi b^2 p}{4}}{\frac{db}{2}} = \frac{\pi b p}{2d}.$$

The average pressure on piston pin bearings is about 2800 lbs. per sq. in. This may seem very high, but is permissible because the rubbing speed is so low. It, however, necessitates the use of a hard bearing metal lining, such as phosphor bronze or special bearing bronze, in the connecting rod, and this in turn calls for the use of a case-hardened piston pin. Hence

$$\frac{\pi b p}{2d} = 2800,$$

$$d = \frac{\pi b p}{5600} = \frac{bp}{1800} \text{ (appr.)}. \quad . \quad . \quad . \quad . \quad (48)$$

The inside diameter of the pin may be determined as follows: The pin is one of the reciprocating parts and should be as light as possible—consistent with the necessary strength. It is always made of a case-hardened steel, plain carbon steel, such as S. A. E. No. 1020, a nickel steel, as Nos. 2315 and 2320, or a chrome nickel steel, such as Nos. 3215 and 3220. With the carbon steels a stress of 20,000 lbs. per sq. in. may safely be allowed, while with alloy steel the stress can be made from 10 to 20 per cent greater. It must be remembered that this is not the regular working stress but the exceptional stress coming on the pin in case of the explosion of the maximum charge that can be introduced into the cylinder.

The piston pin may be regarded as a uniformly loaded beam supported at both ends (Fig. 95). It is made a pretty tight fit in the bosses, and the latter are rendered rigid by a rib connecting them to the piston head. Calling P the total pressure on the pin and l its length between supports, the maximum bending moment is

$$\frac{Pl}{8} = \frac{Pb}{16} = \frac{\pi b^3 p}{64}.$$

The resisting moment of a hollow section of outside diameter d and inside diameter d_1 is

$$M_r = \frac{S\pi}{32}\left(\frac{d^4 - d_i^4}{d}\right).$$

Equating the bending and resisting moments

$$\frac{\pi b^3 p}{64} = \frac{S\pi}{32}\left(\frac{d^4 - d_i^4}{d}\right).$$

$$\frac{db^3 p}{2S} = d^4 - d_i^4,$$

$$d_i^4 = d^4 - \frac{db^3p}{2S},$$

and

$$d_i = \sqrt[4]{d^4 - \frac{db^3p}{2S}}. \quad . \quad . \quad . \quad . \quad . \quad (49)$$

We will now calculate the sizes for a piston pin for a 3 × 4½ in. engine with a compression ratio of 4.5. The maximum explosion pressure with this compression ratio is 450 lbs. per sq. in. The length of bearing is

$$\frac{b}{2} = 1.5 \text{ inches.}$$

The outside diameter of the pin is

$$\frac{bp}{1800} = \frac{3 \times 450}{1800} = 0.75 \text{ inch.}$$

The inside diameter is

$$\sqrt[4]{0.75^4 - \frac{\frac{3}{4} \times 3^3 \times 450}{2 \times 20{,}000}} = 0.54 \text{ inch},$$

say ½ inch. The ends of the piston pin are sometimes tapered out, as there is practically no stress on them, and this permits of a slight reduction of the reciprocating weight. This practice is more prevalent in Europe than here.

Piston pins are generally made of tubular stock, and after being case-hardened they are finished by grinding. This is now generally done in a centerless grinder, in which the work is located between two abrasive wheels rotating in the same direction, a control wheel and a cutting wheel, and is supported by a work rest. When ground in such a machine the work revolves with (at the same peripheral speed as) the control wheel, while the cutting wheel has a somewhat greater peripheral speed. The two wheels are set with their axes not parallel but at an angle to each other, which causes the work to be moved transversely to the wheels and to be ejected from the machine. This finishing operation is a very rapid one.

Piston Bosses—The diameter of the piston bosses must be about 1.4 *d*, where *d* is the outside diameter of the piston pin, and as a rule the bosses are slightly tapered, increasing in diameter toward the piston wall.

Locking Piston Pins—The piston pin must be prevented from moving lengthwise in the piston bosses or "drifting," for if one of its ends shoud come in contact with the cylinder

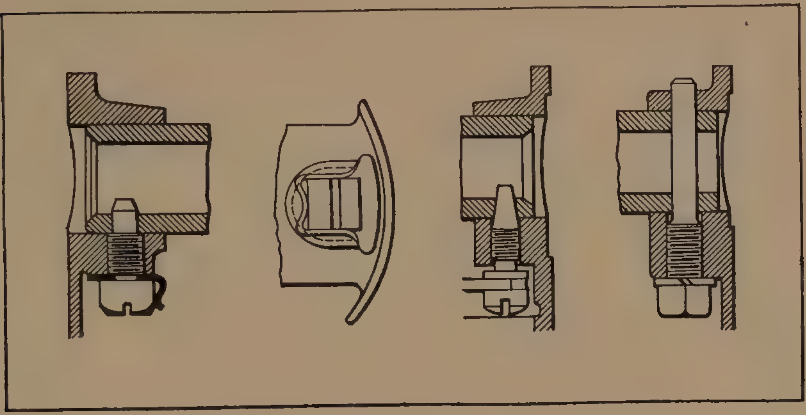


FIG. 96.—METHODS OF LOCKING PIN IN PISTON BOSSES.

it would soon start to score the cylinder bore. In order to minimize the cutting if the pin should come loose, it is well to round off its outer edges when it is being machined.

Where the piston pin is fastened in one of the piston bosses this is now usually done by means of a pin screw, for which a lug is provided on one of the bosses. The locking screw itself is locked either by means of a split pin passed through its head or by a lock washer under a square head. A portion of the edge of the washer follows the inner wall of the piston skirt, and a flap on the washer is turned up against one side of the set screw head. Both of these methods of locking the piston pin are illustrated in Fig. 96.

Piston Pin Fastened in Connecting Rod—Quite a few manufacturers now secure the piston pin in the connecting

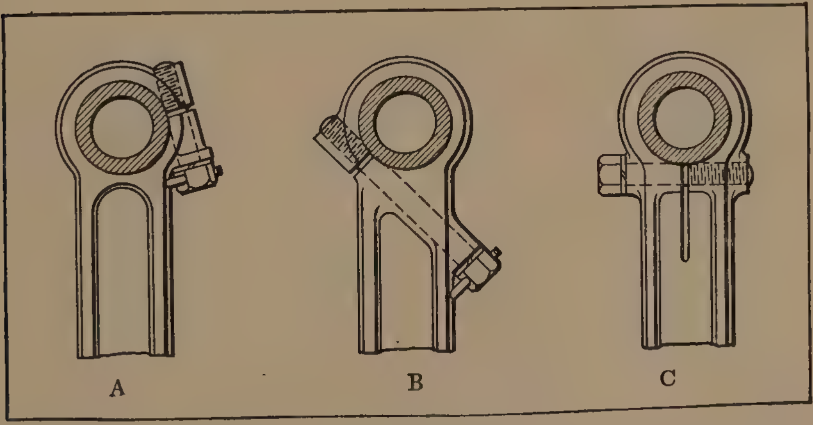


FIG. 97.—METHODS OF CLAMPING PIN IN CONNECTING ROD.

rod small end, so that the pin has bearings in the piston bosses. One advantage of this arrangement is that larger bearing surfaces can be secured. With this construction the length of the bearing surface can be made as much as 0.7 *b*. The pin is generally secured in the connecting rod by means of a clamping screw which passes through a split boss on the hub at the upper end of the rod. The hole for the screw is so located that the screw will pass slightly below the surface of the piston pin, the latter being provided either with a groove running all around it at its middle, or else with a slot which exactly permits the passage of the screw. The split boss for the clamp screw may be located in three different positions relative to the shank of the connecting rod, as shown in Fig. 97 at A, B and C. Arrangements A and B permit of the clamp screw being inserted through the open end of the piston, while arrangement C usually requires a hole in the skirt of the piston through which the screw can be passed. This hole is drilled in that side of the piston which is not under pressure during the power stroke. The screw may be locked in position by means of a spring washer, but it is well to add a split pin, passed through the head, of such a size that the connecting rod will prevent the screw from turning.

Where the piston pin has its bearings in the piston bosses it is not necessary that the latter should be provided with bearing bushings, as cast iron is a good bearing material when the surface speed is low and it is customary to have the pin bear on the bosses direct.

Floating Piston Pin—Some designers fasten the piston pin neither in the piston nor in the connecting rod, but allow it to float in both. Snap rings are then inserted into grooves in the ends of the piston bosses to prevent drifting of the pin (Fig. 98).

The reverse method, that of using a snap ring in a groove at each end of the protruding piston pin, has been used with aluminum pistons, while in Europe the use of bronze or aluminum buttons with short stems fitting into the hollow piston pin enjoys a measure of popularity. It is obvious that the floating construction reduces the danger of a pin seizing, but it is questionable whether more service can be expected from it before repairs become necessary on account of wear and looseness.

Side Pressure on Cylinder Wall—It was shown in a previous chapter that when the momentary effective pressure on the piston per square inch of piston head area is denoted by P_p and the angle which the connecting rod makes with the cylinder axis by ϕ, then the reaction of the cylinder wall against the piston per square inch of piston head area is

$$P_s = P_p \tan \phi. \quad \text{(Equation 31).}$$

The pressure of the piston against the cylinder wall is equal and opposite, and for practical purposes may be represented by the same equation. It is then necessary to find the values of the resultant pressure and of tangent ϕ for different points of the stroke and to multiply corresponding values together. The effective piston pressure is equal to the resultant of the momentary gas pressure in the cylinder and the inertia force. It will be convenient to take points corresponding to the completion of 0.1, 0.2, 0.4, 0.6 and 0.8 of the stroke, beginning at the top end, and determine the values of the resultant pressure and of tan ϕ for these points. We will take the indicator diagram shown in Fig. 16 as a basis. The inertia forces are obtained from Table II, and are added to the corresponding gas pressures. The respective values of the gas pressures and inertia forces, and the resultant pressures, are given in the following table:

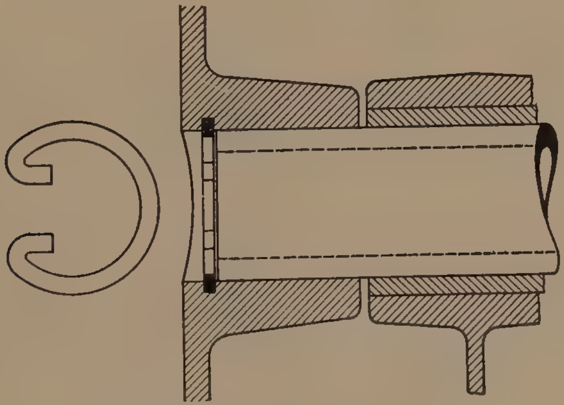


FIG. 98.—FLOATING PISTON PIN.

Point of stroke...	0.1	0.2	0.4	0.6	0.8
Gas pressure	192	151	98	68	52
Inertia force	—95	—65	—11	+33	+63
Resultant ...	97	86	87	101	115

The next problem is to find the values of tangent ϕ for the different points on the stroke. The angle ϕ which the connecting rod makes with the cylinder center line depends upon the angular position θ of the crank and the ratio n of the connecting rod to the length of stroke—

$$\sin\phi = \frac{\sin\theta}{2n} \text{ (equation 23).}$$

We also found (equation 25) that the distance of the piston head from the top end of the stroke is

$$x = \frac{l}{2}(1 - \cos\theta) + nl\left(\frac{\sin^2\theta}{8n^2}\right)$$

$$= \frac{l}{2}\left(1 - \cos\theta + \frac{\sin^2\theta}{4n}\right).$$

Multiplying both sides by $4n$,

$$4nx = \frac{l}{2}(4n - 4n\cos\theta + \sin^2\theta).$$

Substituting $1 - \cos^2\theta$ for $\sin^2\theta$.

$$4nx = \frac{l}{2}(4n - 4n\cos\theta + 1 - \cos^2\theta).$$

Dividing both sides by $\frac{l}{2}$ and transposing,

$$\cos^2\theta + 4n\cos\theta = 4n + 1 - 8n\frac{x}{l}.$$

Completing the square,

$$\cos^2\theta + 4n\cos\theta + 4n^2 = 4n^2 + 4n + 1 - 8n\frac{x}{l}.$$

Extracting roots,

$$\cos\theta + 2n = \sqrt{4n^2 + 4n + 1 - 8n\frac{x}{l}},$$

and, transposing,

$$\cos\theta = \sqrt{4n^2 + 4n + 1 - 8n\frac{x}{l}} - 2n. \quad . \quad . \quad (50)$$

This equation enables us to determine directly the crank angle θ corresponding to any piston position. We then determine from a table of sines and cosines the value of the sine of θ, which we divide by $2n$ (equation 23). This gives us the sine of the connecting rod angle ϕ. Then, from a table of sines, we find the corresponding angle, and finally from a table of tangents, the tangent of this angle. The results of the various steps are given in the following table (for $n = 2$):

Point of stroke.	0.1	0.2	0.4	0.6	0.8
Cos θ	0.837	0.669	0.313	—0.076	—0.507
Sin θ	0.547	0.743	0.950	0.997	0.862
Sin ϕ..........	0.137	0.186	0.237	0.249	0.215
ϕ..........	7° 52′	10° 43′	13° 43′	14° 25′	12° 25′
Tan ϕ	0.138	0.189	0.244	0.257	0.220

Multiplying the values of P_p by those of tan ϕ, we get the following values for the piston pressure on the cylinder wall for different points of the stroke (in pounds per square inch of piston head area):

Point of stroke..........	0.1	0.2	0.4	0.6	0.8
Pressure of cylinder wall..	13.4	16.2	21.2	25.9	25.2

In a similar manner we can determine the side pressures on the cylinder wall during the other strokes.

Allowable Mean Unit Side Thrust—The values here given are the side thrusts per square inch of piston head area. To get the total side thrust we multiply by the piston head area in square inches.

Figure 99 shows the total side thrust on the cylinder wall of a 3½ × 5-inch engine, having reciprocating parts weighing 4.33 lbs., compression and expansion curves like those shown in Fig. 16 and running at 1800 r.p.m. It will be noted that the side thrust changes direction at the beginning and near the middle of each stroke, except the power stroke. The average value of the total side thrust in this engine under the conditions stated is about 110 lbs. Now, the wear on the piston and cylinder wall is proportional to the average side thrust per square inch of bearing surface, consequently the effective bearing surface should be made proportional to the total average side thrust. For the average side thrust per square inch of bearing surface throughout the four strokes we may take 8 to 10 lbs. Hence in the 3½ × 5 inch engine we should need pistons with an effective bearing length of, say,

$$\frac{110}{9 \times 3.5} = 3\tfrac{1}{2} \text{ inches.}$$

In the case of the 3½ × 5 inch piston, if we figure on three piston rings each 3/16 inch wide, do not count the upper land, ¼ inch wide, as effective bearing surface and allow 3/16 inch for width of oil grooves, the total ineffective

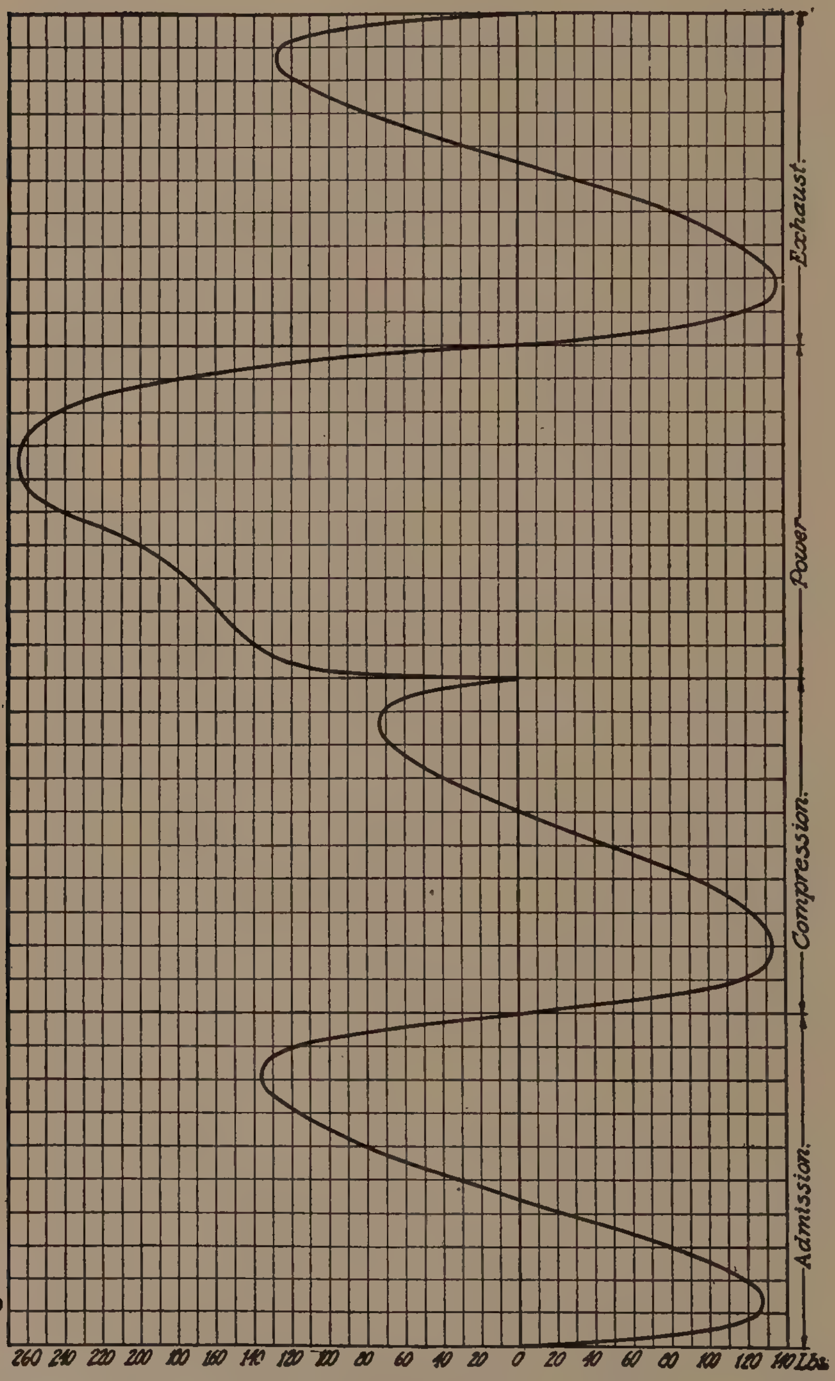


Fig. 99.—Side Thrust on Cylinder Wall of 3½ by 5 In. Engine at 1800 r.p.m.

length will be 1 inch. The total piston length must therefore be made about 4½ inches.

While the method thus described seems to be the most logical for determining the piston length, in practice a short cut is generally employed, by making the total piston length proportional to the diameter.

Pistons for the engines of low-priced passenger cars are often of a length equal to their diameter, while high-grade truck engines sometimes have pistons of a length equal to 1.4 times their diameter. For the same class of service the ratio of piston length to diameter is in a sense an index of the quality of the engine. For heavy duty service, as on trucks, railcars and motor boats, the pistons should be longer. In high-speed engines for passenger cars the need for light reciprocating parts encourages the use of shorter pistons. In the majority of the better grade engines the piston length

$$l_p = 1.1b - 1.25b$$

Distribution of Side Thrust—The component of the connecting rod reaction which results in side pressure on the cylinder wall, passes through the piston pin axis perpendicular to the axis of the cylinder, and is, therefore, a concentrated force. The reaction of the cylinder wall on the piston, on the other hand, is distributed over the entire bearing surface of the piston. If the piston had a bearing surface over its entire length and the piston pin were exactly at the middle of its length, then the reaction would be evenly distributed over the entire bearing surface. But if the piston pin axis were, under these conditions, nearer the head end of the piston, for instance, then the reaction would be greater at the head end than at the open end. The specific pressures at the two ends under these conditions would be inversely proportional to their distances from the piston pin axis, and the pressure would decrease uniformly from the head end to the open end. This effect is illustrated in Fig. 100. The grooves disturb the distribution of the pressure somewhat, and their presence must be taken into account. What is wanted, of course, is uniform distribution of pressure over the entire bearing length of the piston.

Best Location of Pin—It is now customary to regard the lands of the ring belt as of no value as bearing surface and to locate the piston pin axis substantially midway between the lower ring groove above the piston bosses and the open end of the piston. In the earlier editions of this book a formula was given taking account of the effect of the friction between the piston and cylinder wall (which during the power stroke increases the pressure at the lower end and decreases it at the top end), but by reason of the exceedingly low friction coefficient in a well-lubricated engine, this effect is of no practical consequence and can be disregarded.

Piston Materials—Up to about 1914 the common material for pistons was gray iron, but about that time a demand for a decrease in weight of reciprocating parts arose and aluminum alloys were introduced for the purpose. At first considerable difficulties were encountered as a result of the much greater coefficient of heat expansion of aluminum alloy as compared with cast iron. These difficulties have now been overcome, and aluminum pistons are used to a considerable extent in this country and even more, proportionately, abroad. While the specific gravity of aluminum alloy is around 3, as compared with about 7.5 for cast iron, the saving in weight due to the use of aluminum for pistons is much less than might be expected from these figures. This is due to the fact that the aluminum alloy has less strength, particularly at high temperatures, and with it the wall thickness must be made greater. Recently a magnesium alloy has come into limited use for piston manufacture, and as magnesium has a specific gravity of only 1.8 as compared with 3 for aluminum alloy, it permits of exceedingly light pistons.

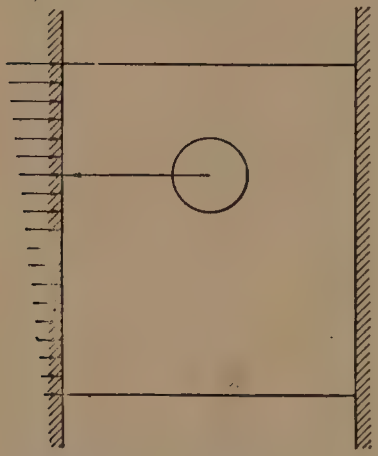


FIG. 100.—DISTRIBUTION OF PRESSURE WITH UNSYMMETRICAL LOAD.

While the demand for lighter pistons arose in the first place in connection with high-speed engines, designers of moderate speed truck and tractor engines also have come to realize the advantages of light reciprocating parts, and the old style cast iron piston with a skirt ⅛ in. or more thick has passed entirely out of use. A keen rivalry has set in between the champions of the cast iron and the aluminum alloy pistons, and in consequence the design of cast iron pistons has been so refined that in some cases this piston is now made lighter than an aluminum piston of the same diameter designed in accordance with established rules.

In the very light cast iron pistons as now used in high speed engines the thickness of wall sections is chiefly a matter of foundry limitations. The thickness of the skirt is generally made 1/16 in., with a total tolerance of perhaps 0.020 in., and very little more metal is allowed at the bottom of the ring grooves. Moreover, to further lighten the piston certain sections of the skirt on the sides of the bosses are sometimes cut away, giving what is known as a skeleton piston.

Metal Mold Iron Castings—Some of these light iron pistons are cast in metal molds in a machine invented by D. H. Meloche of the Holley Carburetor Co. The molds are made of cast iron in two parts that are held together by spring pressure. In order to prevent too severe chilling, a wash composed of powdered fireclay, waterglass and water is applied while the mold is hot, so that the water evaporates and leaves a durable coating of mineral matter. After each pouring a further coating is applied by means of a smoky acetylene flame. Dry sand cores are used with this process. The piston castings remain in the mold for 47 seconds and are then ejected. They are said to be true to dimensions within 0.005 in. and to be free from sand, scale, blowholes and hard spots.

Three hundred to four hundred pistons are turned out per hour by one machine. The castings are normalized at 1550° F. for twenty minutes and after cooling are tumbled and sand-blasted. They show a Brinell hardness of 176-185. When the machine is first started the molds must be preheated, while in continuous operation they readily maintain a sufficiently high temperature. Between successive pourings the mold is blown out with compressed air, which also cools it. Pistons cast by this process are in use on several large production passenger cars.

Allowance for Heat Expansion—The average American practice with cast iron pistons is to allow 3 thousandths of an inch per inch of cylinder bore at the head end, and ¾ thousandth on the skirt. The piston diameter on the top land, therefore, should be

$$d_h = 0.997b$$

and the piston diameter at the skirt,

$$d_b = 0.99925b$$

The diameters on the second and third lands are made intermediary between those of the top land and the skirt.

Number and Location of Rings—Practice is about evenly divided between the use of three and of four rings. If a light, compact engine is desired, it will be advisable to limit the number of rings to three, while if lightness is not such an important consideration, it will be better to use four. Generally, all of the rings are placed on that part of the piston above the piston pin, but occasionally a ring is placed near the lower or open end. This is an inheritance from stationary gas engine practice. It is sometimes claimed that the object of this ring is to aid in the distribution of the oil, but the original purpose for which the ring was placed at the lower end was to have it over-travel the bottom end of the cylinder bore, and thus prevent the forming of a ridge where the travel of the lowest ring ends. It is usual to cut two or three circular grooves in the outer surfaces of the piston near the lower end for the distribution of the oil.

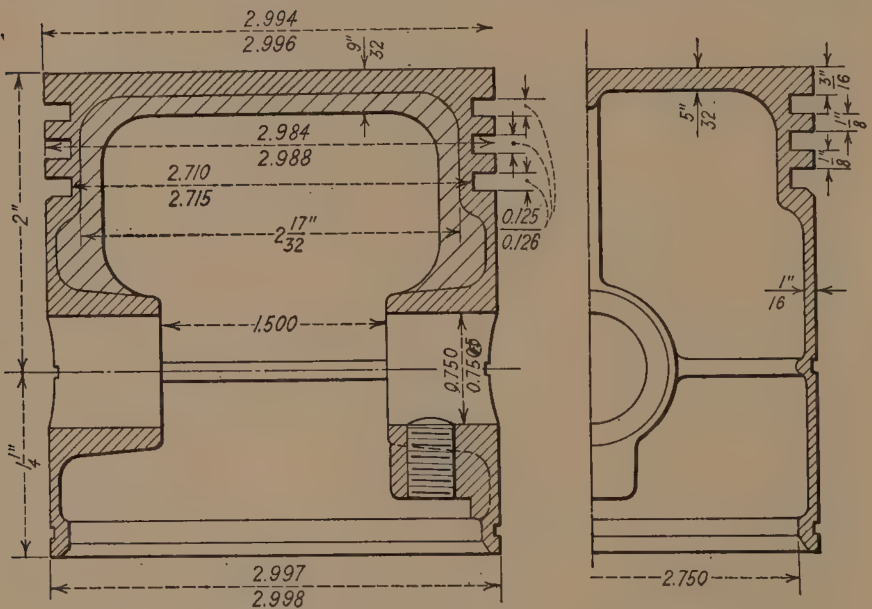


Fig. 101.—Light Cast Iron Piston for 3-In. Cylinder.

Sample Design—In as much as the wall thickness of light cast iron pistons depends more upon foundry limitations than upon working stresses, the design of such pistons will be illustrated by an example, which will serve the purpose of the formula given in connection with other parts. Such a piston for an engine of 3-in. bore is illustrated in Fig. 101. At the open end there is an inside flange to provide the stiffness necessary for machining. The piston bosses, too, are made comparatively light, the average diameter being 1 1/16 in. with a piston pin hole of a nominal diameter of ¾ in. At the bottom of the ring grooves the wall is only 5/64 in. thick, while the piston head, which is plane, is 5/32 in. thick.

Near the top of the piston, grooves are provided for three ⅛-in. rings with 3/32-in. lands between them, and a top land 3/16 in. wide. The skirt of the piston is finish-ground to a diameter of 2.998 in., while the top land is ground to 2.995 in. Usually, a tolerance of 0.001 is allowed on these dimensions. For instance, the top diameter would be specified as 2.994-2.996 and the open end diameter as 2.997-2.998.

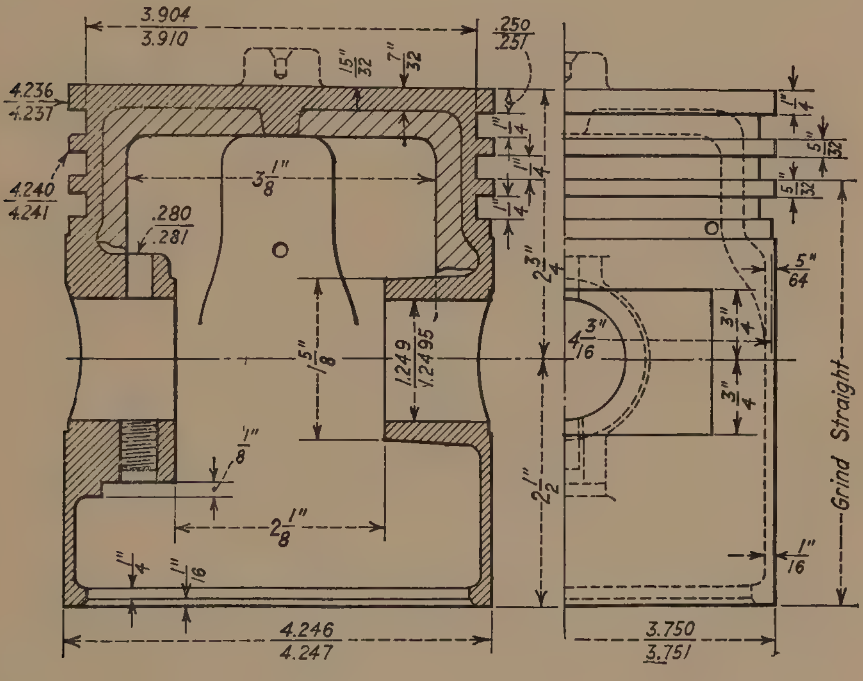


Fig. 102.—Cast Iron Piston for 4¼-In. Cylinder.

A cast iron truck piston design is shown in Fig. 102. This has a nominal diameter of 4¼ in. and length of 5¼ in. The finish diameter for the top end is 4.236-4.237 in., and for the skirt 4.246-4.247. The skirt is 5/64 in. thick while the thickness of metal below the ring grooves is 9/64 in. and the head is 7/32 in. thick. The head is supported by eight evenly spaced ribs ¼ in. high, of which two connect to each piston boss. The hole through the piston bosses is of $1\frac{1}{4}$ in. nominal diameter and only one-half of a thousandth inch tolerance is allowed on this. The spaces around the ends of the piston pin holes are relieved by grinding eccentrically $1\frac{1}{2}$ in. wide and to a maximum depth of 1/64 in. the eccentricity being so chosen as to carry each relief one quarter way around the piston.

Oil Pumping—A good deal of trouble has been experienced at various times from "oil pumping," i.e., the sucking of oil into the combustion chamber when the engine runs on small throttle. This trouble was most acute when aluminum pistons first came into practical use and had to be given a large clearance in order to eliminate danger of seizing. When engines with such pistons run under small load they are comparatively cool and the pistons, consequently, are not fully expanded, so that the excess clearance is not taken up.

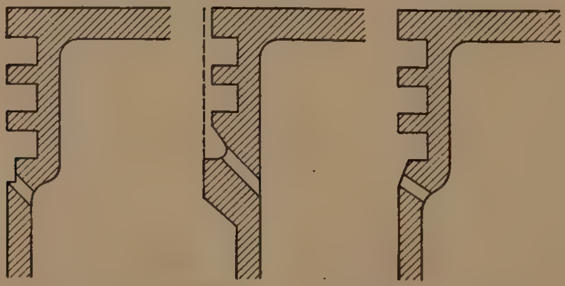


FIG. 103.—THREE DESIGNS OF OIL RETURN GROOVES AND HOLES.

To prevent such oil pumping the piston should have a groove cut around its circumference just below the bottom ring, at the bottom of which groove is drilled a number of equally spaced holes (about 3/32 in.) through the piston wall. The lower piston ring then acts as a scraping ring, the oil collecting in the groove below it and flowing through the drill holes back into the crankcase. Three designs of oil grooves and return holes are shown in Fig. 103.

Slipper Piston—For very high speed work, and particularly for racing engines, use is sometimes made of a design known as the slipper piston, which is due to Harry Ricardo. As will be seen by referring to Fig. 104, those portions of the skirt at right angles to the plane of connecting rod swing are cut away. The object is to reduce the friction between piston and cylinder wall. The cutaway portions of the skirt have practically no value as bearing surface, as the pressure between piston and cylinder wall is parallel to them. However, the clearance between these surfaces and the cylinder bore is filled with oil, and as the piston reciprocates in the cylinder, the film of oil is "sheared," which consumes energy and therefore causes a loss of power. At very high speed this loss of power is considerable, especially if the engine is operating at low temperature so the oil is comparatively viscous. Other advantages claimed for this type of piston are as follows: Piston thrust is carried directly to the piston bosses; the two slippers are made of different sizes, in proportion to the average thrusts against the cylinder wall during the two down and the two up strokes, respectively; the piston is inherently light, yet its bosses are supported rigidly. These pistons are used to a certain extent in the small bore stock engines of English small cars. American engineers, while recognizing its advantages, have hesitated to adopt the slipper piston for stock engines, because they believe that the elimination of a large fraction of the oil film would make it more difficult to hold compression.

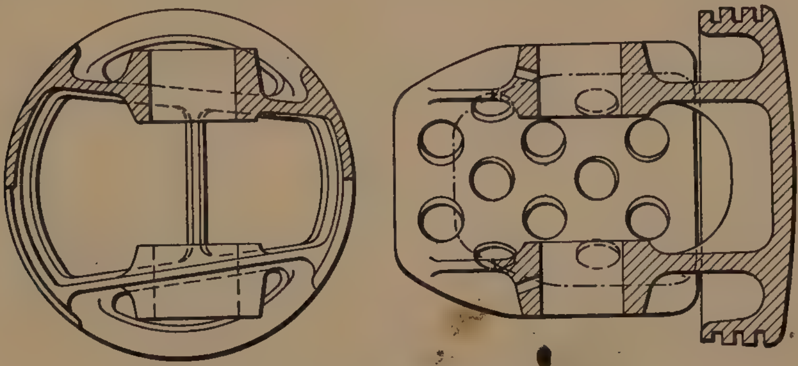


FIG. 104.—RICARDO SLIPPER-TYPE PISTON.

Aluminum Alloys for Pistons—Many different alloys are used for pistons, but they all seem to have one thing in common—a small magnesium content. They may therefore be regarded as related to an alloy developed during the early part of this century and which was known as magnalium. It contained an appreciable proportion of magnesium and its specific gravity was only 2.5. These light alloys differ materially from cast iron in their physical properties, a fact which must be taken into account in designing pistons to be cast in them. Their fusing points (1,100-1,200° F.) are very much lower, consequently care must be taken that the temperature of the piston head will not become too high, for at temperatures approaching the melting point the alloys lose a great deal of their strength. In racing engines fitted with aluminum alloy pistons it has happened that the piston heads have collapsed as a result of attaining a very high temperature while operating under full load.

Aluminum has a coefficient of heat expansion (0.00001234) substantially twice that of cast iron, and this has been the cause of the greatest difficulties which have been encountered with alloy pistons. Obviously, in order to prevent seizing of the pistons when the engine runs continuously under heavy load, a greater clearance must be allowed between piston and cylinder wall. Fortunately the thermal conductivity of aluminum is much greater than that of cast iron, hence the heat absorbed by the piston head is transferred to the skirt much more rapidly and danger of overheating of the head is thereby reduced. If too much clearance is allowed the piston is apt to develop side slap while cold.

The Y-Alloy—An aluminum alloy suitable for piston castings was developed in England by the National Physical Laboratory and is known as the Y-alloy. It contains 4 per cent of copper; 2 per cent of nickel and 1½ per cent of magnesium. At 32° F. this alloy has a tensile strength of 30,000 lbs. per sq. in., and test specimens show an appreciable amount of elongation, which increases rapidly above 550° F.

In preparing this alloy certain precautions are necessary. The magnesium must be added just before pouring, preferably in the form of fairly large pieces of pure metal, which should be immediately pushed below the surface of the aluminum, so that they can dissolve without coming in contact with the air. If added at an early stage of the melting operation, considerable magnesium is likely to be lost. Nickel is added as follows: A hardener or rich nickel alloy (about 20 per cent) is first prepared, and this is added in the proper proportion to the molten aluminum.

Castings of this alloy often show numerous pin holes, but tests have shown that these do not materially weaken the metal. Determinations of the coefficient of thermal conductivity gave values ranging from 0.35 for the sand-cast metal at 212° F. to 0.41 for the annealed metal at 575° F. compared with 0.102 for cast iron as cast, at 212°, to 0.113 for annealed cast iron at 575° F.

This alloy is responsive to heat treatment. For instance, a 1-in. diameter chill-cast test specimen as cast showed a tensile strength of 28,000 lbs. per sq. in. at atmospheric temperature, but when heated in a nitrate (salt) bath to 900°

for 48 hours, quenched in water and aged, the tensile strength was 36,000 lbs. and the elongation 2.25 per cent in 2 in.

Lynite—Metal Mold Castings—Most of the aluminum alloy pistons used in this country are cast from alloys known as Lynite, made by the Aluminum Co. of America. Lynite alloys for pistons contain from 8 to 10 per cent of copper, 0.5 per cent of magnesium and 1 to 1.5 per cent of iron.

Aluminum pistons may be cast either in sand or in metal molds. The metal mold process permits of casting the pistons very close to size and reduces the amount of machine work required and the waste of material in shavings. It also materially increases the hardness of the alloy, thus improving the wearing properties.

An incidental advantage of aluminum pistons is that carbon deposits do not adhere to them as well as to cast iron pistons. It is also claimed that owing to the greater heat conductivity of aluminum the compression can be raised by as much as 10 per cent when such pistons are used.

Clearance Allowance in Aluminum Pistons—When aluminum pistons were first introduced they were usually made of the same general form as iron pistons, except that the sections were made heavier, or deeper, and more numerous ribs were provided to make up for the lesser strength of the alloy. It was found that in order to prevent seizing of the pistons under heavy load, the clearance had to be made much larger than in the case of cast iron pistons. The usual clearance for such "solid walled" alloy pistons was 0.002 inch per inch bore on the skirt, 0.010 inch more on the top land and proportionate amounts more on the inermediate lands. It was generally conceded that in view of the great importance of having the clearance just right, that is, no greater than absolutely required to prevent seizing of the pistons under load at high speed, the clearance problem must be solved experimentally for each engine and piston design.

The comparatively large clearances required with these early aluminum pistons led to a great deal of trouble from piston slap and oil pumping, and the way in which the problems arising therefrom were dealt with may be briefly reviewed.

Preventing Piston Slap—Some manufacturers recommended making the pistons comparatively long, say 1.3 times the diameter, and this undoubtedly gave some relief. The plan of making the pistons long also commended itself because these aluminum alloys do not have the same resistance to wear as cast iron pistons, and lengthening them reduced the unit pressure on the wearing surface and thus tended to increase piston life.

The Packard Motor Car Co. patented a scheme whereby the piston pin axis is offset from the piston axis, which is intended to cause the piston to "cock" in the cylinder and thus to contact with both sides of the bore at all times. A. L. Nelson suggested the use of steel struts, provided with enlarged, knurled heads cast into enlargements of the walls, to hold apart or brace the pressure sides, these being partly separated from the piston pin sides by vertical slots. Pistons made under his patents and using struts of Invar, a high nickel content alloy steel of very low heat expansion, are now used by Chrysler and others. Patents have also been taken out on the process of finishing the piston skirt to an elliptical form by moving the grinder wheel head back and forth as in cam grinding, or by expanding the piston along one diameter when finishing it.

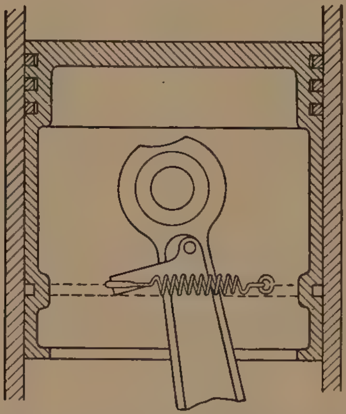


Fig. 105 (left).—Crossley Method of Preventing Piston Slap.

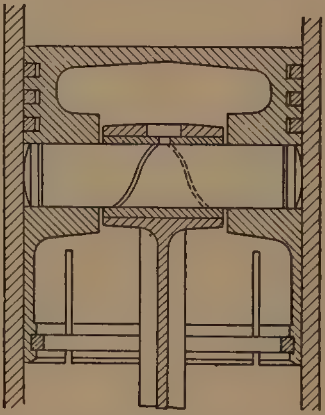


Fig. 106 (right).—Humber Slitted Piston with Expanding Ring.

One British firm (Crossley), in order to prevent piston slap, used strong coiled springs under tension, one end of which was anchored to a bracket secured to the connecting rod shank and the other passed through a hole in the skirt of the piston (Fig. 105). Another (Humber) splits the piston skirt lengthwise from the open end and inserts a spring ring in a groove cut on the inside near the open end, which tends to expand the skirt into contact with the cylinder wall all around (Fig. 106).

Split Skirt Piston—Most of the troubles with aluminum pistons were solved by the introduction of the split skirt or constant clearance type of piston. One such design is illustrated in Fig. 107. The skirt is partly cut off from the ring belt by horizontal slots in the wall extending about half-way around. Reliefs around the ends of the piston bosses are cast in. A 1/32-in. saw slot extends the whole length of the skirt at a slight angle to the vertical so as to prevent non-uniform wear of the cylinder wall. The provision of this slot permits of limiting the clearance on the skirt to 0.001 inch per inch of diameter. Owing to the fact that the skirt is not a complete cylinder, that part of the piston evidently cannot seize in the cylinder bore. Its partial separation from the ring belt reduces the flow of heat from the piston head to the skirt. There are two substantial ribs between the piston head and the piston bosses. The fact that the piston bosses are nearly symmetrically located with respect to the supporting wall tends to give them a more rigid support.

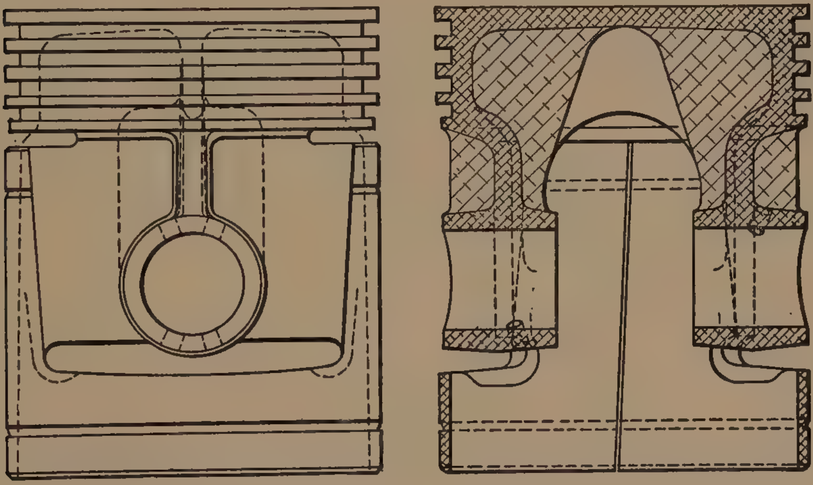


FIG. 107.—SPLIT-SKIRT TYPE OF ALUMINUM PISTON.

It will be noted that there is a large slot at the bottom of each relief, which serves to return excess oil to the crankcase. The "vertical" slot is cut on the side of the piston which is free of pressure during the power stroke. The wall thickness of the skirt tapers from ⅛ in. at the top end to 3/32 in. at the open end. The average thickness of the wall below the ring grooves is about 9/64 in., while the thickness of the head increases from 3/16 at the center to 7/32 near the edge. These pistons, which are for an engine of 3¼-in. bore, are held to a weight limit when finished, of 14 oz. and the castings are given a heat treatment to give a Brinell hardness of 90-110.

The general type of piston design illustrated by Fig. 107, which is in use on many American makes of engine, is due to the engineers of Aluminum Manufacturers, Inc.

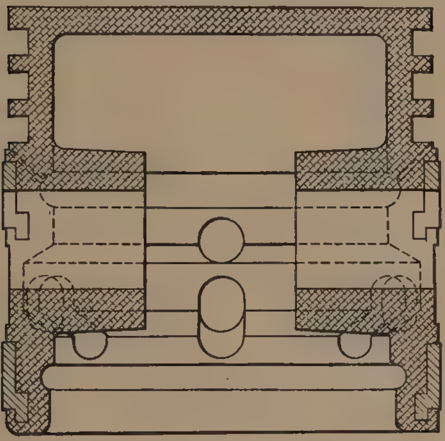
In some other "split skirt" pistons the skirt is separated from the ring belt by a saw cut all around and is supported from the head by pillars or stays which also support the piston bosses. In one British design the pillars are inclined toward the piston axis so that the effect of the gas pressure on the piston head and the reaction between piston pin and bosses is to expand the skirt.

Composite Pistons—In order to combine the lightness of aluminum with the good wearing qualities of cast iron, Fiat casts two thin cast iron rings with an anchoring flange in the aluminum piston (Fig. 108). With the same object in view, Marmon casts the head and bosses of aluminum and the skirt of cast iron, and then machines the parts and bolts them together, by means of four studs, the nuts on which are secured by washers with turned-up flaps. This permits of finishing the skirt on the inside over almost the whole length. Two dowel pins are provided at the joint to insure continued accurate alignment of the two parts. This piston is illustrated in Fig. 109.

Magnesium Pistons—Previous to the War magnesium was produced mainly in Germany, and it seems that the first pistons of magnesium alloy were made by the Griesheim Elektron Chemical Works. The specific gravity of magnesium is only 1.8, or about two-thirds that of aluminum, and since magnesium is added to aluminum to improve its bearing qualities, it promised not only to permit of cutting down the weight materially below the minimum obtainable with aluminum, but also to ensure good bearing (and wearing) conditions. Special attention was called to this new piston material by a contest for light alloy pistons held in Germany in 1921 under the auspices of the Minister for Transport. In connection with this competition extended laboratory and road tests were made, and pistons of Elektron metal made by Hellmuth Hirth were awarded the first prize. These pistons were of a special design, with unusually thick sections at the head end.

As regards experience with magnesium pistons in practice, G. Ruarg, General Manager of the Berlin General Omnibus Co., reports in the Yearbook of the German Automobile Manufacturers' Association for 1925, that with magnesium pistons 25,000 miles had been covered with one bus without trouble of any kind and without undue wear on piston and cylinder walls. The engine of this bus had run 18-20 hours per day without shut-down, and the bus had covered 125 miles and had been stopped and accelerated some 900 times per day. Up to the end of 1924 the Elektron Works had delivered more than 100,000 pistons of magnesium alloy.

In respect to heat conductivity magnesium ranks slightly behind aluminum (134 to 175) when a comparison is made on the usual basis of bodies of equal length and equal cross-section; but if the section of magnesium is increased so that the weight is the same as in the case of aluminum, the former has an advantage of about 15 per cent. Magnesium can be machined at the highest cutting speeds known to industry, nearly 2000 ft. p m., without showing a tendency to tear or "smear." The metal also takes a good, sharp thread.



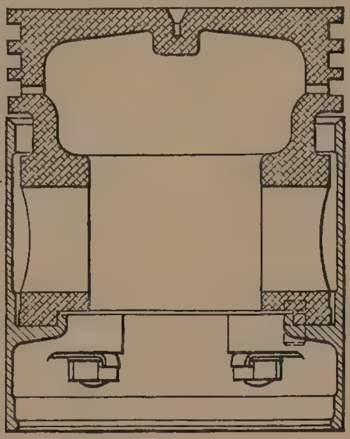


FIG. 108 (LEFT).—FIAT ALUMINUM PISTON WITH CAST IRON WEAR RINGS.

FIG. 109 (RIGHT).—MARMON COMPOSITE PISTON WITH CAST IRON SKIRT.

The coefficient of heat expansion for pure cast magnesium is 0.0000142; for cast aluminum alloyed with 8 per cent of copper, 0.0000128; for cast aluminum alloyed with 12 per cent of silicon, 0.0000102; for cast aluminum alloyed with 12 per cent of copper, 0.0000146, and for gray cast iron, 0.00000558. When magnesium is chill-cast it improves with respect to both tensile strength and ductility (elongation). The shrinkage allowance for patterns is the same for magnesium as for cast iron (⅛ in. per foot) and about 20 per cent greater for magnesium-aluminum alloys and aluminum-copper alloys.

Magnesium piston alloy has a specific gravity of 1.81, a yield point of 20,000-24,000 lbs. per sq. in.; a tensile strength of 36,000-40,000 lbs. per sq. in., an elongation of 15-18 per cent and a Brinell hardness of 46-50.

In this country pistons are made of a magnesium alloy known as Dowmetal by the Dow Chemical Co. of Midland, Mich.

Manufacture of Pistons—Since the walls of the piston castings are very thin, they are often hard in spots, and for this reason the castings are first annealed. They are brought to a good red heat (about 1050° F.) in an annealing furnace and are then allowed to cool slowly in the furnace for about two hours. After annealing the castings are sand-blasted to remove the core sand and scale. Sometimes the annealing operation is dispensed with and the castings instead are allowed to "age," preferably after the rough-turning.

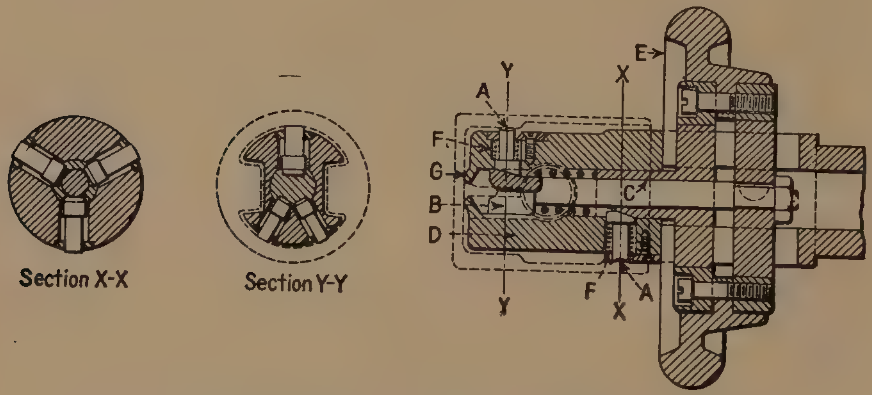


FIG. 110.—EXPANDING ARBOR FOR TURNING PISTONS.

The first machining operations on a piston of the conventional type consist of center drilling and reaming the little boss generally provided on the head, facing the open end and boring out and chamfering. The following operations generally are located from the center hole at the head end and the finished face and flange of the open end. A large center or cone is forced into the open end, with a driving pin or projection engaging with the piston bosses. For these operations the piston must be very accurately centered with relation to its inner surface, and an alternate method of holding it on the turret lathes, designed specially with this end in view, makes use of an arbor with adjusting pins as illustrated in Fig. 110. The six adjusting pins A are actuated by tapered seats on the sliding spindle B and sleeve C, each of which has projections passing through slots in the body D. These projections are threaded right and left hand, respectively, to engage with nuts shown inserted in handwheel E, thus conveying a telescoping action to spindle B and sleeve C. The adjusting pins are returned to their lowest position by springs F.

The arbor is held in the turret and projects as little as possible. The piston is slipped on the arbor until the underside of the head comes against stop G. In case there is a cross-rib in the piston head the stop should be slotted to clear it. The pins are then expanded by means of handwheel E to hold the piston rigidly, and the work is carried forward to a floating chuck which grasps it by the head and which in turn is secured to the back plate before the expanding pins are released.

Rough-turning of the skirt, rough facing of the head and roughing out the ring grooves is generally accomplished in one setting in an automatic lathe. The remarkable speed at which these operations are performed may be judged from the fact that when working on an aluminum alloy piston of 3⅜-in. diameter, 60 pistons are handled per hour. This operation is followed by corresponding finishing operations. The width of the ring grooves has to be kept within very close limits (.005-001 inch), and with alloy pistons a sizing or burnishing finishing operation is sometimes employed, which gives a very smooth and slightly hardened surface. The bottom corners of the ring grooves must be sharp, and some manufacturers cut a narrow, shallow recess in each corner. The head may be finish-faced and the top land reduced in diameter in one operation.

It is even possible to perform all of the machining operations on the circumference and on the head in one setting, and a tool layout designed to accomplish this in a Fay automatic lathe is shown in Fig. 111. The carriage turns the outside diameter ready to grind, while the back arm faces the head, roughs the ring grooves, cuts the oil grooves and forms the outside diameter of the lands, ready for grinding. The carriage then rocks forward and finishes the ring grooves.

Machining Aluminum—A few words may here be said on the machining of aluminum. The tool should have about 15° side rake; it is fed in about 0.025 inch per revolution and a cutting sped of 400-500 ft. p. m. is recommended. Best results are obtained if the cutting edge is set slightly above the axis of the work. As a lubricant for cutting aluminum, a mixture of one part kerosene and two parts mineral lard oil is used to advantage.

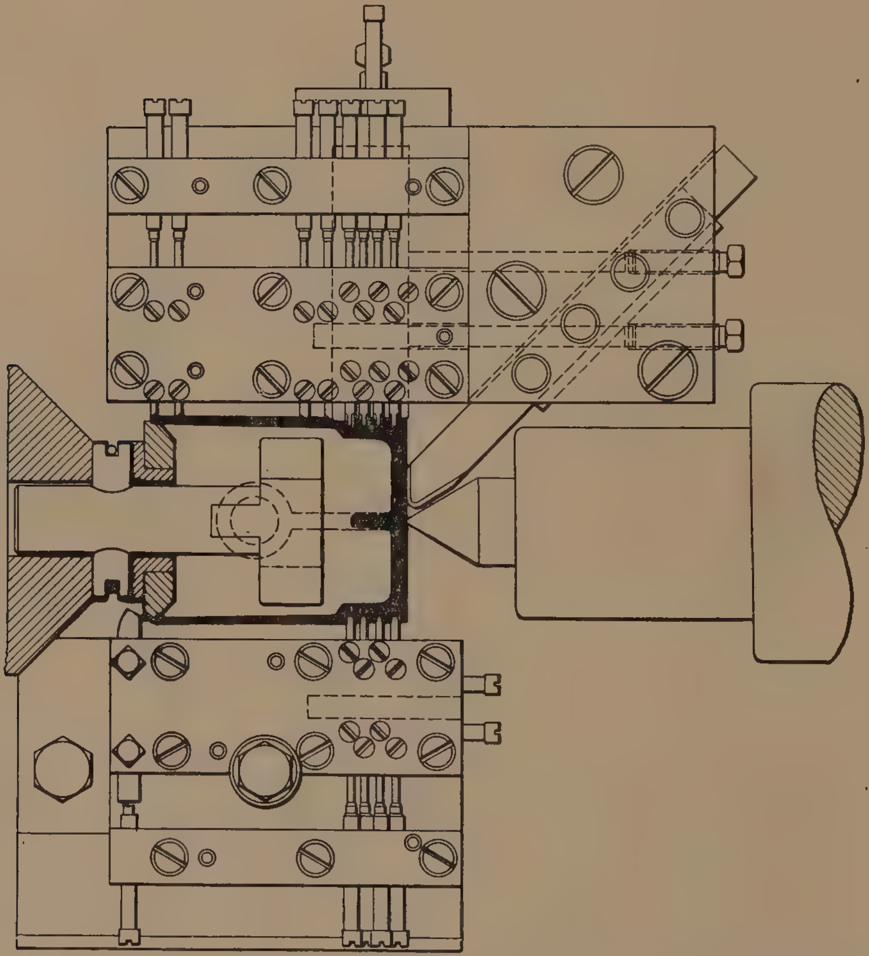


FIG. 111.—TOOL LAY-OUT FOR MACHINING PISTONS IN FAY LATHE.

Boring Piston Pin Holes—Next the holes in the piston bosses are bored out and finished, which is a very important operation, since these holes must be very closely at right angles to the piston axis. If they should be off by even a small amount the piston would "cock" in the cylinder bore and be apt to stick or bind, or its lower edge would act as an oil scraper. On larger pistons the holes are cored, but in smaller sizes the bosses are frequently cast solid. One method with solid-cast bosses consists in first drilling them out, then boring them with a single cutter boring tool, next machine-reaming with a floating reamer, and finally hand-reaming them. For the boring operations the piston is held in a jig with a plane surface against which the faced head is clamped. The bosses are aligned with the bushings in the fixture by means of sliding and spring-supported V blocks engaging both of them. In the case of cored bosses alignment may be assured by inserting a pilot bar through the guide bushing of a jig of the type illustrated in Fig. 112, into the cored hole. Piston pin holes must be kept to within very close limits, the tolerances in the case of one aluminum alloy piston with 7/8 in. pin being 0.0003 in.

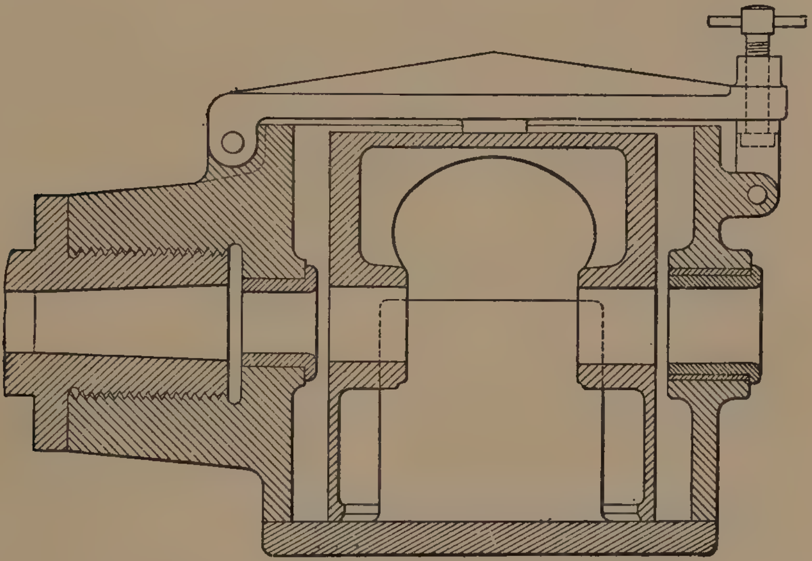


Fig. 112.—Jig for Drilling Piston Bosses.

For drilling and reaming the piston bosses, special double-ended machines have been evolved, and a similar machine is available for grinding these small holes. An advantage claimed for the latter is that it is not necessary to feed the grinding wheel all through the hole, and that the "bell-mouthing" tendency is eliminated. A double-ended drill press used for this purpose is shown in Fig. 113.

Many pistons have a machined relief over the ends of the bosses and this is generally obtained by grinding as illustrated in Fig. 114. The piston is held in a fixture by means of which it can be swung around an axis eccentric to its own axis.

Grinding the Skirt—The skirt of the piston is practically always finished by grinding in two operations, the piston being held between the cone and center, with the driving pin engaging the bosses. In the rough-grinding operation about 0.012 in. stock is removed all around, and in the finish grind, 0.0015 in. With aluminum pistons, owing to their greater heat expansion, it has been found a good plan to hold the temperature of the coolant during the finish-grinding operation within narrow limits.

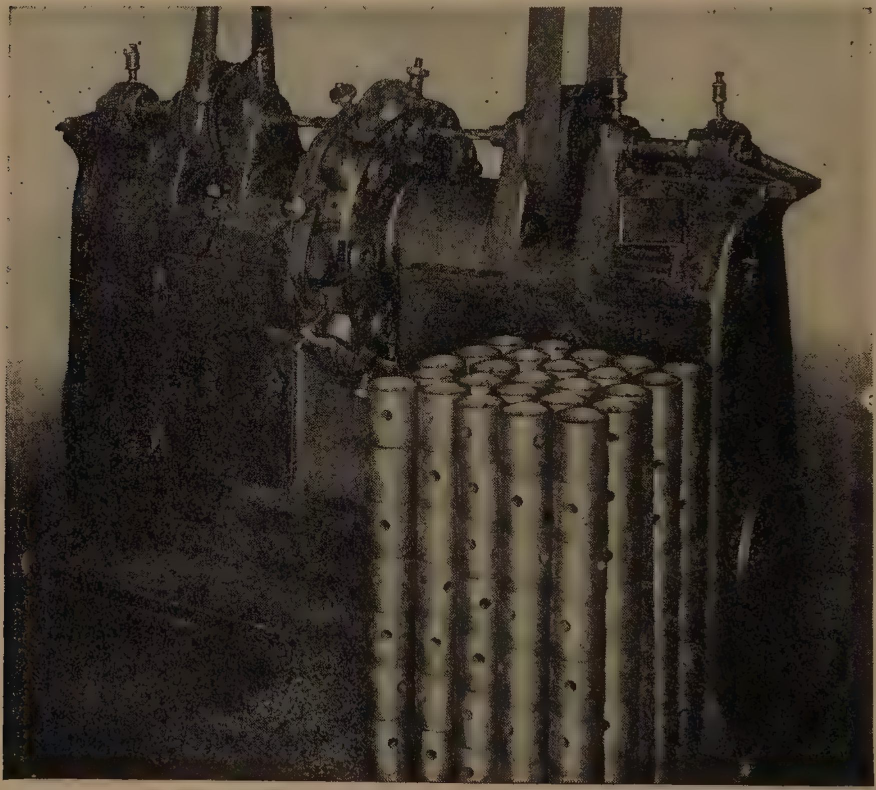


FIG. 113.—DOUBLE ENDED DRILL PRESS FOR DRILLING PISTON BOSSES.

With the modern type of split-skirt piston there are a number of saw-slotting operations which come near the end of the machining process. Finally the center boss on the head is removed and sometimes the head is ground on a surface grinder so as to obtain a smooth surface which does not hold carbon so readily and absorbs less heat.

Pistons are checked for diameter by means of dial gauges, and sometimes are sorted for selective assembly according to diameter. They are also checked by weighing, and on some of the light pistons for high-speed engines the weight tolerance is set at 0.01 lb. On the larger pistons for truck engines the tolerance is ½ oz. One of the most important checking operations is that for alignment of piston pin holes with the face at the open end (which latter is assumed

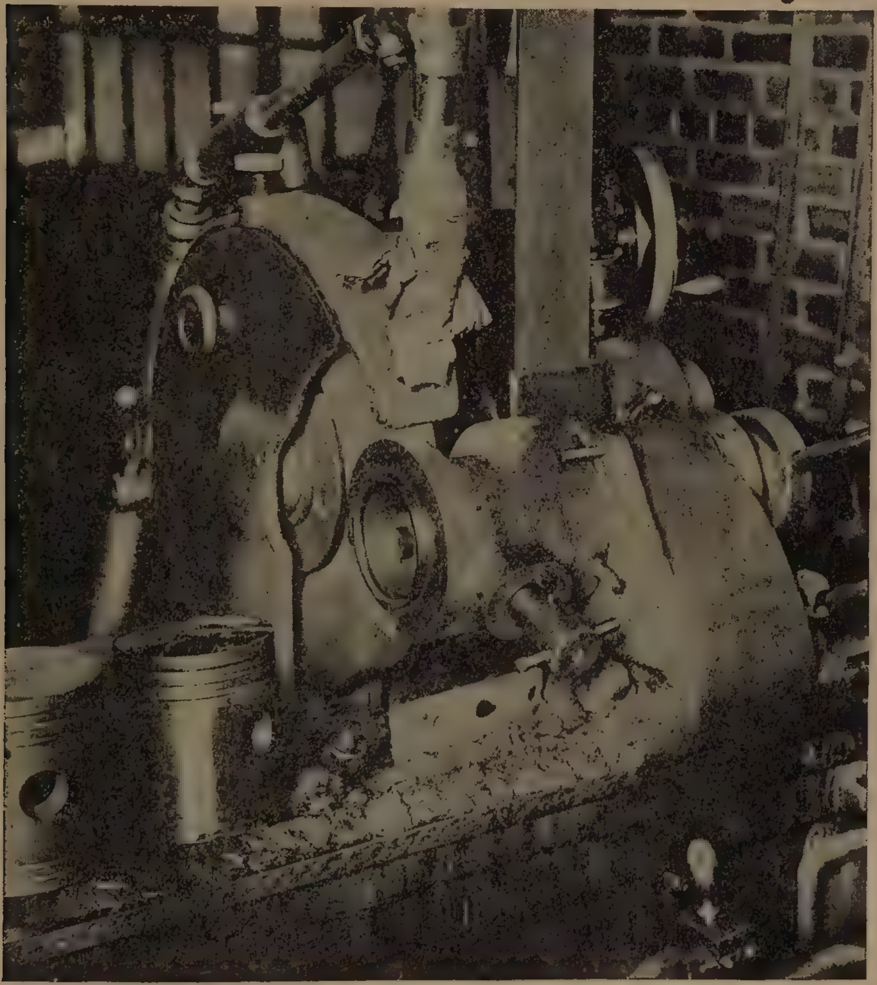


FIG. 114.—GRINDING RELIEF ON PISTONS.

to be in alignment with the piston axis.) The piston is placed on a surface plate, and a bar is passed through the piston pin holes. By means of a dial gauge the height of the top of the bar where it projects from the piston on one side is checked against the other side, the piston being swung halfway around for the purpose. The tolerance on mis-alignment is generally held to 0.001 inch per inch.

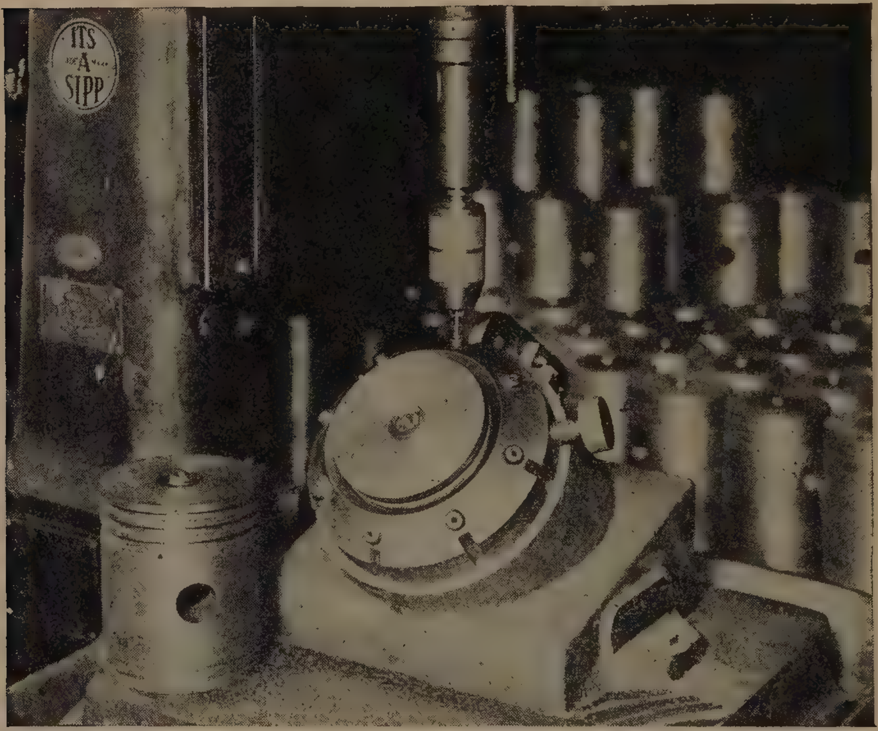


FIG. 115.—DRILLING OIL-RETURN HOLES IN PISTON IN INDEXING JIG.

CHAPTER VIII

Offset Cylinders

Object of Offsetting—The side thrust of the piston against the cylinder wall is the cause of both wear of the piston and cylinder and of frictional loss, and any means of reducing it is therefore doubly advantageous. It was shown in a previous chapter that this side thrust is proportional to the effective pressure on the piston and to the tangent of the connecting rod angle. The effective pressure on the piston cannot well be reduced, as upon it depends the power developed by the engine, so the only chance lies in reducing the value of tangent ϕ (that is, of the angle ϕ itself). The most obvious way of accomplishing this is by making the connecting rod longer. Old textbooks on the steam engine dwell at considerable length on the ideal engine with a connecting rod of infinite length, in which, among other things, there is no side thrust on the cylinder wall. In practice, however, the length of the connecting rod is limited, since the height of the engine, and consequently its bulk and weight, increase rapidly with this length. There is, however, a method of arranging the moving parts so that with a fixed length of connecting rod the connecting rod angle will be relatively small when the pressure on the piston is high, and proportionately greater when the pressure on the piston is low. This consists in offsetting the cylinder axis from the vertical plane through the crank shaft axis toward the side on which the crank arm moves during the power stroke. The angularity of the connecting rod for any position on the power stroke, except at the very ends of the stroke, is then less than it is for the corresponding position in an engine without offset cylinders—a symmetrical engine.

Offsetting of the cylinders (or of the crankshaft) has some rather peculiar effects on the functions of the engine. In the first place, the piston is not at the ends of its stroke when the crank arm is in the vertical position; it reaches the ends of the stroke a moment later, when the crank arm and connecting rod are in line with each other. Besides, the two dead center positions are not 180° apart, as they are in a symmetrical engine, but slightly less, as clearly shown in Fig. 116. The effective or actual length of the stroke, instead of being equal to twice the length of the crank arm, is slightly greater than this. The connecting rod is parallel with the cylinder axis some time after the top vertical position of the crank has been passed and the same time before the bottom vertical position is reached. The two points for which the connecting rod angle becomes zero both fall in the downward stroke. The piston is not at the same distance from the top end, say, of the stroke, when the crank arm is at the same angular distance from its upper vertical position on opposite sides of the center, respectively, as is the case in the symmetrical engine.

Length of Stroke and Dead Center Positions—Referring to Fig. 116, when the crank and connecting rod are in the upper dead center position, the sine of the crank angle

$$\sin\theta = \frac{f}{\frac{l}{2} + nl} = \frac{f}{l(n + \frac{1}{2})}, \quad \dots \quad (51)$$

and when the piston is in the bottom dead center position

$$\sin\theta_1 = -\frac{f}{nl - \frac{l}{2}} = -\frac{f}{l(n - \frac{1}{2})}. \quad \dots \quad (52)$$

In our particular example of a 4 by 5-in. engine with an offset of $b/4 = 1$ inch

$$\sin\theta = \frac{1}{5 \times 2.5} = 0.08,$$

and

$$\theta = 4° 35';$$

also

$$\sin\theta_1 = -\frac{1}{5 \times 1.5} = -0.133,$$

and

$$\theta_1 = 187° 40'.$$

If we assume that the crank motion is uniform, which is very nearly the case, it follows that the mean piston speed is slightly less during the down stroke than during the up stroke, since the crank travels through a greater angular distance during the down stroke than during the up stroke.

Referring to Fig. 116, it can easily be seen that the actual length of stroke is

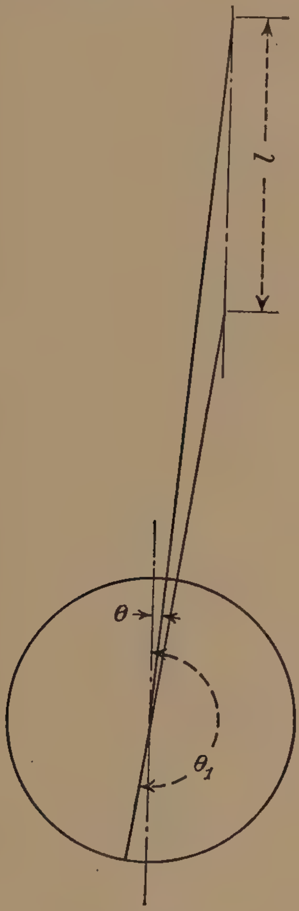


Fig. 116.

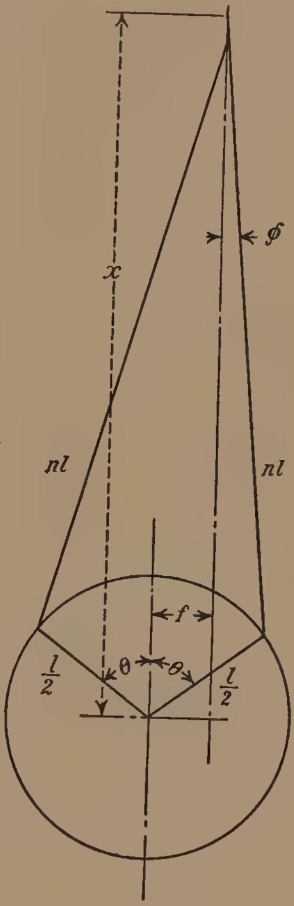


Fig. 117.

$$S = \sqrt{\left(nl + \frac{l}{2}\right)^2 - f^2} - \sqrt{\left(nl - \frac{l}{2}\right)^2 - f^2}$$

$$= \sqrt{l^2(n + \tfrac{1}{2})^2 - f^2} - \sqrt{l^2(n - \tfrac{1}{2})^2 - f^2}. \quad . \quad (53)$$

For an engine of 5-inch nominal stroke with 10-inch connecting rod and 1-inch offset this gives

$$s = 5.027 \text{ inches,}$$

while for the same engine with an offset of 0.666 inch (one-sixth the bore) it gives

$$s = 5.012 \text{ inches}$$

The main object of offsetting the cylinder being a reduction in the side thrust of the piston against the cylinder wall, we will investigate the variation of this side thrust in an offset cylinder engine during the power stroke and compression stroke, respectively. We will also investigate the effect of offsetting on the timing of the engine and on the balance of the reciprocating parts. To this end it is necessary to develop equations for the piston position, piston speed and piston acceleration corresponding to any crank angle, the same as was done for the symmetrical engine.

Piston Speed and Acceleration—Referring to Fig. 117, let f represent the amount of the offset. Let the length of the connecting rod be denoted by ln and the length of the crank arm by $l/2$ (though it should be pointed out that in this case l stands for the nominal length of stroke and not the actual). The use of the same symbols as in the previous discussion has the advantage that the equations developed should reduce to the corresponding equations for the symmetrical engine if f is made 0, which will be a check on the accuracy of the work. The crank and connecting rod angles are again represented by θ and ϕ, respectively. As regards the momentary piston position, it is more convenient in this case to represent this by the vertical distance between the piston pin axis and the crankshaft axis, rather than by the distance between the momentary position of the piston pin axis and its highest position.

Let x represent the vertical distance between the piston pin axis and the crankshaft axis. Then (Fig. 117),

$$x = nl \cos \phi + \frac{l}{2} \cos \theta,$$

and

$$nl \sin \phi + f = \frac{l}{2} \sin \theta.$$

This latter relation holds for any point of the crank circle.

It follows that

$$\sin \phi = \frac{\frac{l}{2} \sin \theta - f}{nl} = \frac{\sin \theta}{2n} - \frac{f}{nl}. \quad . \quad . \quad . \quad (54)$$

But

$$\cos\phi = \sqrt{1-\sin^2\phi} = \sqrt{1-\left(\frac{\sin\theta}{2n}-\frac{f}{nl}\right)^2}.$$

Hence,

$$x = nl\sqrt{1-\left(\frac{\sin\theta}{2n}-\frac{f}{nl}\right)^2}+\frac{l}{2}\cos\theta. \quad . \quad . \quad (55)$$

In order to get rid of the radical expression we add to it

$$\tfrac{1}{4}\left(\frac{\sin\theta}{2n}-\frac{f}{nl}\right)^4,$$

which makes it a perfect square and does not introduce an appreciable error. Extracting the root, we get

$$x = nl - \frac{nl}{2}\left(\frac{\sin\theta}{2n}-\frac{f}{nl}\right)^2+\frac{l}{2}\cos\theta. \quad . \quad . \quad . \quad (56)$$

If now we differentiate this expression with respect to time we get the momentary piston speed

$$v = \frac{dx}{dt} = \left[-nl\left(\frac{\sin\theta}{2n}-\frac{f}{nl}\right)\frac{\cos\theta}{2n}-\frac{l}{2}\sin\theta\right]\frac{d\theta}{dt}.$$

But

$$\frac{d\theta}{dt} = \frac{\pi N}{30}\ \textit{radians per second},$$

hence,

$$v = -\frac{\pi N}{30}\left(\frac{l}{4n}\sin\theta\cos\theta-\frac{f}{2n}\cos\theta+\frac{l}{2}\sin\theta\right).$$

Dividing by 12 to reduce to feet per second, and multiplying the expression in parentheses by 2, and dividing the coefficient of this expression by the same number,

$$v = -\frac{\pi N}{720}\left(\frac{l}{2n}\sin\theta\cos\theta-\frac{f}{n}\cos\theta+l\sin\theta\right)\ \text{ft. per sec.} \quad (57)$$

When $f = 0$ this equation becomes identical with equation (26), giving the momentary piston speed for a symmetrical engine.

In order to find the piston acceleration we differentiate equation (57), which gives

$$dv = \frac{\pi N}{720}\left[\frac{l}{2n}(\cos^2\theta-\sin^2\theta)+\frac{f}{n}\sin\theta+l\cos\theta\right]d\theta,$$

and remembering that

$$\frac{d\theta}{dt} = \frac{\pi N}{30},$$

and that

$$\cos^2 \theta - \sin^2 \theta = \cos 2\theta,$$

we get

$$a = \frac{dv}{dt} = -\frac{\pi^2 N^2}{21{,}600}\left(\frac{l}{2n} \cos 2\theta + \frac{f}{n} \sin \theta + l \cos \theta\right)$$

$$= -\frac{\pi^2 N^2 l}{21{,}600}\left(\frac{1}{2n} \cos 2\theta + \frac{f}{nl} \sin \theta + \cos \theta\right). \quad . \quad . \quad (58)$$

This equation becomes identical with equation (28) (which gives the piston acceleration for a symmetrical engine) when the offset f is zero, except that it is preceded by the minus sign, which is due to the fact that in this case we are considering the motion of the piston pin axis relative to the crankshaft axis, and this is opposite to the motion of the piston pin relative to its highest position, which we considered in the former case.

If W represents the weight of the reciprocating parts, then the inertia force in the offset engine may be represented by the equation

$$F_a = \frac{WlN^2}{70{,}480}\left(\frac{1}{2n} \cos 2\theta + \frac{f}{nl} \sin \theta + \cos \theta\right). \quad . \quad (59)$$

Side Thrust on Cylinder Wall—The length of the offset employed in practice lies generally within the limits of one-quarter the bore and one-sixth the bore. American practice rather favors an offset of one-sixth, while European practice favors an offset of one-fourth. In determining the side thrust for different points of the up stroke and the down stroke we must first determine the crank angle corresponding to different points on both strokes. The deduction of an equation giving this angle or a function of it directly involves considerable difficulty, and the best method is to proceed backwards, assuming different crank angles and then determining the corresponding point on the stroke by means of equation (55). From the results thus obtained a curve like that shown in Fig. 118 can be drawn. It is, of course, also possible to determine the points for this curve graphically without calculation, by laying off the connecting rod and crank arm for different angular positions of the latter. From Fig. 118 we can readily determine the crank angle corresponding to any point on the stroke. The connecting rod angle is next determined by means of equation (54) and the effective piston pressure from the standard diagram. Multiplying corresponding values of the connecting rod angle and the effective piston pressure together, we obtain the side thrust on the cylinder wall.

In this way the curves in Fig. 119 were plotted. The calculations were based on the assumption of a cylinder of 4-in. bore by 5-in. stroke. The full line curve gives the side thrust of the piston on the cylinder wall per square inch of piston head area for an offset equal to one-sixth the bore, and the dotted line the corresponding values for an offset equal to one-quarter the bore. In a 4 × 5 in. engine one-sixth of the bore corresponds to two-fifteenths of the stroke and one-quarter of the bore to one-fifth the stroke. In the figure the offsets are given in terms of the stroke so as to make the diagram applicable to all sizes of engines. For the sake of comparison, the side thrust diagram for a symmetrical engine obtained under the same conditions is also drawn in in dashed lines. Since we are not so much concerned with what side of the cylinder the piston presses against, as with the total pressure with which it bears against the cylinder walls, all side thrusts are plotted on the same side of the base line, irrespective of their direction.

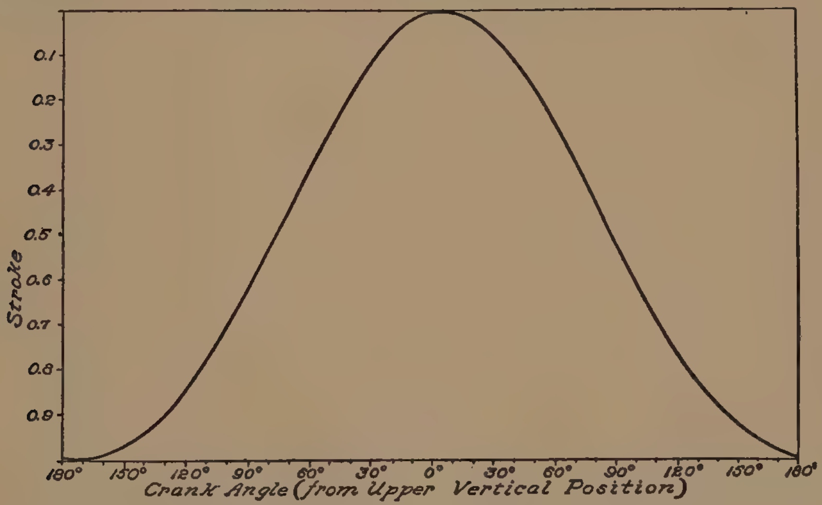


FIG. 118.—CRANK ANGLE-STROKE CURVE FOR ENGINE WITH AN OFFSET EQUAL TO ONE-FIFTH THE STROKE.

By reference to Fig. 117 it can readily be seen that in an offset engine the connecting rod angle is comparatively small throughout the power stroke, and comparatively large throughout the compression stroke. For this reason the side thrust on the cylinder wall in an offset engine is less than in a symmetrical engine during the power stroke and greater than in a symmetrical engine during the compression stroke. This is clearly shown in Fig. 119. In a symmetrical engine

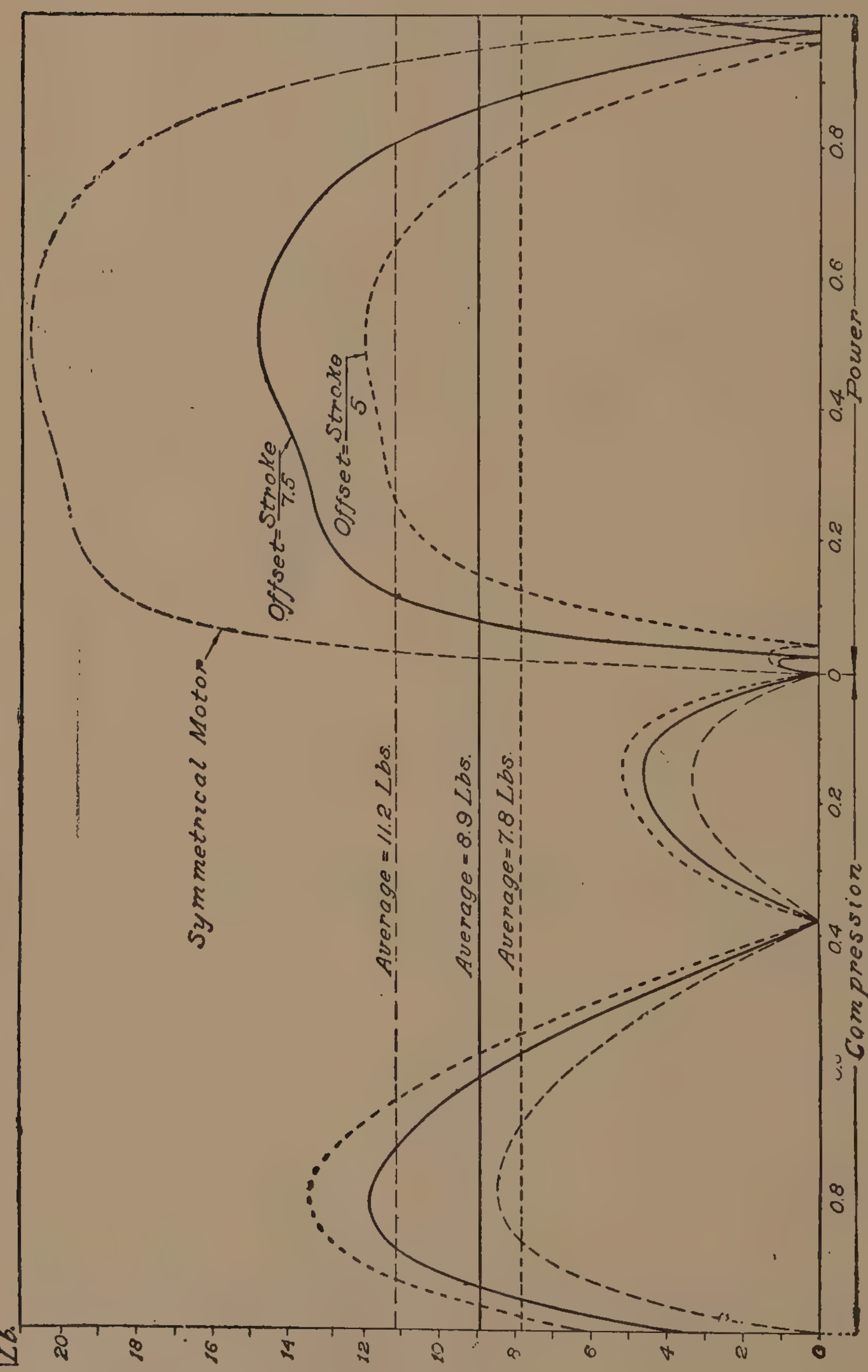


FIG. 119.—SIDE THRUST ON WALL OF $3\frac{1}{2}$ BY 5-IN. CYLINDER AT 1800 R.P.M.

the side thrust becomes zero when the crank is in the two vertical positions. In an offset engine, on the other hand, when the crank is in its lowest position (represented in the diagram by the left-hand co-ordinate axis), the connecting rod makes an appreciable angle with the cylinder axis. During the first part of the compression stroke, the greater the offset the greater the side thrust, because the greater the angularity of the connecting rod. The side thrust becomes zero in all three types of engine when the piston is slightly less than four-tenths of the stroke from its top position, because, in the diagram worked from the inertia and gas pressure then neutralize each other so that the effective piston pressure is zero. At the end of the compression stroke the side thrust is again zero, because at this point of the diagram the inertia and compression pressure again balance each other. From this point on the side thrust in the symmetrical engine increases rapidly; in the offset cylinder engine, on the other hand, it increases for a moment and then drops back to zero again, which it attains when the connecting rod center coincides with the cylinder axis; from that point on the side thrust increases in about the same manner as in the symmetrical engine, but decreases to zero again a moment before the crank reaches the lowest position, when the connecting rod is again in line with the cylinder axis. It will be seen from Fig. 119 that under the conditions here assumed, the mean side thrust on the cylinder wall during the compression and power strokes is 11.2 lbs. per sq. in. of piston head area in a symmetrical engine; 8.9 lbs. in an engine with an offset equal to two fifteenths of the stroke, and 7.8 lbs. in an engine with an offset equal to one-fifth the stroke. Offsetting the cylinder by an amount equal to two-fifteenths the stroke results therefore in a reduction of 20.5 per cent in the side thrust on the cylinder wall during the compression and power strokes, and offsetting by an amount equal to one-fifth the stroke in a reduction of about 30 per cent.

Effect on Timing—The valves should begin to open and should close and the spark in an engine should occur when the piston is at particular points of its stroke, and since any particular point of the stroke corresponds to a different crank angle in an offset than in a symmetrical engine, it is plain that these functions should occur at different angular positions of the crank in an offset than in a symmetrical engine. If the designer knows of a valve timing which gives satisfaction in a symmetrical engine and wants to apply the same system in an offset engine, he should first determine, by means of equation (24), the piston positions corresponding to the different crank angles used in the timing system in the symmetrical engine, and then determine by means of Fig. 118 (or a similar curve if the offset is more or less than one-fifth the stroke) the crank angles corresponding to these piston positions in an offset engine.

One of the peculiar effects of offsetting in a four-cylinder engine is that all four pistons travel in the same direction—downward—for a moment after two have passed the top dead center.

Effect on Balance—An inertia force curve for a 4 × 5 inch engine with an offset of one inch, reciprocating parts weighing 0.55 pound per square inch of piston head area

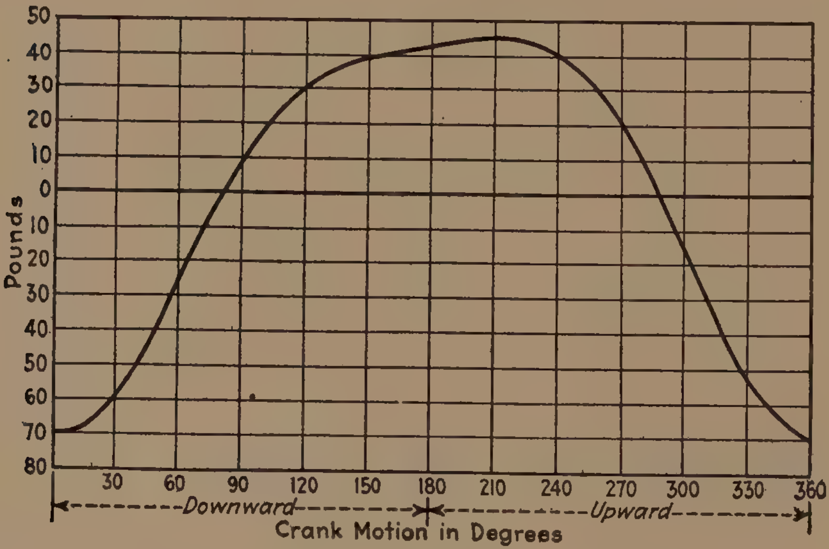


FIG. 120.—INERTIA FORCE DIAGRAM FOR A CYLINDER WITH AN OFFSET EQUAL TO ONE-FIFTH THE STROKE.

and operating at 1000 feet piston speed per minute, is shown in Fig. 120. The data from which this curve was drawn were calculated by means of equation (59). It will be noticed that the curve is considerably distorted. When the crank is in the vertical positions the inertia forces are exactly the same as in a corresponding symmetrical engine. The inertia reaches both of its maxima during the upward stroke.

In Fig. 121 one-half of this inertia curve is shifted 180° with relation to the other. This represents the conditions in a four-cylinder engine in which the motion of one pair of cranks is advanced 180° with relation to that of the other pair. The two portions of the inertia curve are drawn in in dotted lines and their resultant is drawn in in a full line. This resultant is exactly the same as is obtained in the case of a corresponding symmetrical engine; consequently, offsetting of the cylinder has absolutely no effect on the unbalanced inertia force in an engine. That this is so can also easily be proven analytically.

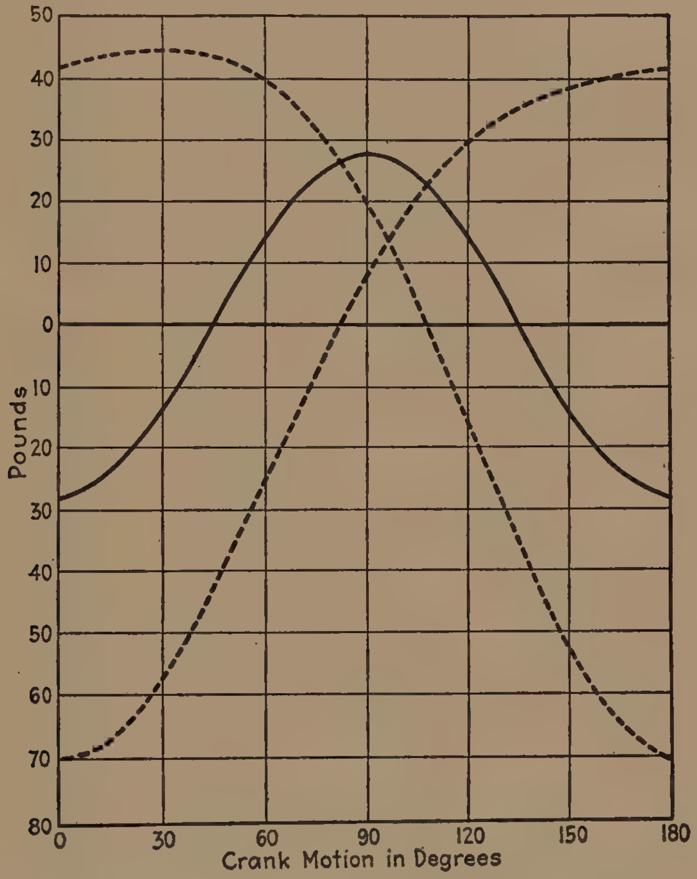


Fig. 121.—Inertia Force Diagram for a Four-Cylinder Offset Engine.

In a four-cylinder engine there are two sets of moving masses separated by 180° of crank motion. The inertia of one set may be represented by the equation

$$F_a = \frac{WlN^2}{70{,}480}\left(\frac{1}{2n}\cos 2\theta + \frac{f}{nl}\sin\theta + \cos\theta\right),$$

and that of the other by

$$F_{a_1} = \frac{WlN^2}{70{,}480}\left[\frac{1}{2n}\cos 2(\theta + 180^\circ) + \frac{f}{nl}\sin(\theta + 180^\circ) + \cos(\theta + 180^\circ)\right].$$

The resultant unbalanced inertia force is the sum of these two expressions.

But

$$\sin\theta = -\sin(\theta + 180^\circ)$$

and

$$\cos\theta = -\cos(\theta + 180^\circ),$$

hence the terms involving the factors $\sin\theta$ and $\cos\theta$ cancel out in the sum. On the other hand,

$$\cos 2(\theta + 180^\circ) = \cos(2\theta + 360^\circ) = \cos 2\theta.$$

Hence

$$F_a + F_{a_1} = \frac{WlN^2}{70{,}480}\left(\frac{1}{n}\cos 2\theta\right). \quad . \quad . \quad . \quad . \quad (60)$$

This is the equation of the unbalanced inertia force. It will be seen that it does not involve the factor f, and the same result is obtained if the expression for the unbalanced inertia force in a four-cylinder symmetrical engine is made the basis of the discussion.

With engines of comparatively long stroke offsetting of the cylinders involves some difficulty, because there usually is not sufficient clearance for the connecting rod during the power stroke. This difficulty can be overcome by casting the bottom end of the cylinder with a sort of slot, into which the connecting rod can swing.

In the above discussion a low-speed engine was purposely chosen as an example because offsetting is undoubtedly most advantageous in engines of that class. This is due to the fact that in a high-speed engine there is much side thrust on the cylinder walls due to the inertia, and since the average of this is the same for both up and down strokes the average value of the side thrust due to it for the two strokes cannot be reduced by offsetting. For instance, in the 3½ × 5 engine, which we have been considering previously, the average side thrusts per square inch of piston head area at 1800 r.p.m. during the four strokes of a cycle are as follows:

Inlet	8.9 lbs.
Compression	7.5 lbs.
Power	20.4 lbs.
Exhaust	8.9 lbs.

It will readily be seen that if by offsetting we reduced the average side thrust during the power stroke in the ratio of, say, 4:3, and increased the average side thrust during the compression stroke in the inverse ratio, the result on the total average for the four strokes would be very small. In fact, it would be reduced only from 11.4 to 10.8 lbs.

CHAPTER IX

The Crankshaft

One of the most important parts of the gasoline engine is the crankshaft. It is an expensive part, requiring a considerable amount of accurate machine work to be done upon it, and it is subjected to very severe stresses, consequently it is liable to fail in service unless properly dimensioned and made of a suitable grade of material properly treated.

Crankshaft Material—Medium carbon steel is the material most extensively used. The blanks from which the crankshafts are machined are generally produced by the drop forging process. This process requires several heatings to a good red heat, which impairs the physical qualities of the steel, and the latter have to be restored by suitable heat treatment, consisting of annealing, quenching and reheating. The carbon steel generally used for crankshafts has approximately the following composition and physical properties (after heat treatment):

Chemical Composition

		Per Cent
Carbon		0.45
Manganese		0.70
Silicon	not over	0.18
Sulphur	not over	0.04
Phosphorus	not over	0.04

Mechanical Properties

Tensile strength	110,000 lbs. per sq. in.
Elastic limit	75,000 lbs. per sq. in.
Elongation in 2 inches	20 per cent
Contraction of area	50 per cent
Brinell hardness	225-245

A suitable heat treatment for steel of this character is as follows: The forgings are first heated to a temperature of about 1500° F. and quenched in oil. They are then reheated to 1450° and allowed to anneal (cool slowly) in lime. They are heated again to 1450° and quenched in oil, and are then drawn to about 1000°. The drawing temperature may be varied more or less, a lower drawing temperature giving a stronger shaft and a higher one a tougher or more ductile shaft.

In some of the higher grade automobile engines, chrome-nickel steel (S. A. E. No. 3140) is used for the crankshaft. The composition and the (approximate) mechanical properties of this steel after heat treatment are as follows:

Chemical Composition

Carbon	0.35-0.45
Manganese	0.50-0.80
Phosphorus ... not over	0.04
Sulphur ... not over	0.45
Nickel	1.00-1.50
Chromium	0.45-0.75

Mechanical Properties

Tensile strength	140,000 lbs. per sq. in.
Elastic limit	115,000 lbs. per sq. in.
Elongation in 2 inches	18 per cent
Contraction of area	53 per cent
Brinell hardness	280-300

The heat treatment for this steel consists in first normalizing it, that is, heating to between 1625 and 1725° F. and allowing to cool freely in the air; then raising to between 1475° and 1525° and quenching in oil, and finally drawing at about 950°.

Other materials used for crankshafts include chrome-vanadium and chrome-molybdenum steels. All of these ternary alloys have excellent mechanical properties, the elastic limit (when suitably heat-treated) exceeding 100,000 lbs. per sq. in. In a few instances (Franklin and International Motors in this country) the crankshafts are case-hardened on the bearing surfaces. This necessitates the use of steels of low carbon content, either plain carbon or alloy (such as S. A. E. Steel No. 3120). With this material a glass-hard surface is obtained which is practically free from wear if the crankshaft is supported in the usual white metal bearings.

In determining the required dimensions of a crankshaft such mechanical properties as the tensile strength are generally disregarded. The reason for this is that one of the chief requirements in a crankshaft is stiffness or rigidity. This depends upon the coefficient of elasticity of the material, which is substantially the same for all grades of steel (in the neighborhood of 30,000,000). The only sound reason for using alloy steel is that it is harder and therefore resists wear better than carbon steel. Nickel particularly has the effect of hardening steel when added in moderate proportions.

Types of Crankshaft—A crankshaft is composed of the crankpins, crank arms, crank journals and driving ends. As a rule, crankshafts are forged in a single piece, but occasionally they are built up. Built-up crankshafts are used in small single- and double-cylinder motorcycle engines. The enclosed flywheels of these engines take the place of the crank arms, the crankpin and crank journals being bolted to the flywheels, which latter are cast with solid webs. The built-up construction also has advantages when it is desired to support the crankshaft in three or more ball bearings, as with a one-piece shaft all intermediate bearings would have to be stripped over the crank arms and therefore would have to be made extra-ordinarily large.

A crankpin, together with the two crank arms on opposite sides of it, is frequently referred to as a "throw." In some crankshafts there is only a single throw between a pair of crank journals or supporting bearings; in other crankshafts there are two throws between crank journals; in others three, and in still others four. The dimensions of the crankpins naturally are largely determined by their function as a bearing. The factor that limits the load capacity of a bearing is its ability to get rid of the frictional heat generated, which ability is approximately proportional to its projected area or the product of its length by its diameter. The heat generated varies with the total bearing load, the surface speed and the friction coefficient. From these relations the conclusion might be drawn that if the surface speed were decreased by decreasing the diameter of the bearing, the unit pressure could be increased, and within certain limits this holds true. There is, however, one other factor to be taken into consideration, and that is crankshaft distortion. If under the loads upon it the crankshaft flexes inordinately, the journals will not press evenly against the bearings over their whole surface, and, in consequence, the load and the resulting heating may become excessive in spots, which may result in the destruction of the bearing. This explains why it is sometimes possible to even reduce the length of the bearing when the diameter is increased. The crankshaft becomes more rigid and the loads on its bearings therefore are distributed more nearly uniformly over their entire surfaces.

Balance Weights—In a single-cylinder crankshaft the centrifugal force on the crank arms, crankpin and part of the connecting rod forms an unbalanced rotating force which would cause a great deal of vibration if no means were provided for properly balancing it. Therefore, balance weights are applied to the crank arms, as pointed out in the chapter on Balancing of Engines. The method of calculating the proper balance weight was there given. The balance weight should be such that the centrifugal force acting on it is equal to the centrifugal force acting on the crank arms, crankpin and half of the connecting rod, plus one half of the maximum primary inertia force acting on the reciprocating parts.

In a high-speed engine the balance weights are preferably forged integral with the crank arms. If made separate they must be very securely applied, since the stresses on the fastenings due to the centrifugal force at ''racing'' speeds are very considerable, and the loosening of one of the weights would be sure to do serious damage. Alloy steel bolts or studs should preferably be used.

In a double-cylinder opposed engine the crankshaft is always made with two throws set at 180° relative to each other. One set of reciprocating parts then always moves opposite to the other set, and at exactly the same speed, so that the reciprocating parts are perfectly balanced, except for the fact that the two sets are not quite in line with each other. The rotating parts are also very nearly balanced, since the centrifugal force acting on one throw is equal and opposite in direction to the centrifugal force acting on the other throw, and there is only a small rotating couple due to the centrifugal forces acting at the ends of a beam equal in length to the distance between the centers of the two. This rotating couple can be balanced by applying balance weights to the two short crank arms.

In a four-cylinder vertical engine the four throws are always in the same plane, the two outer throws being on the same side of the crankshaft axis, and the two inner throws on the opposite side. The centrifugal force acting on each throw is a radial rotating force which may be considered to act at the center of that throw. The centrifugal forces F on the two outer throws (Fig. 122) naturally are always in the same plane, and as they are equal, their resultant R is a force equal to twice that acting on the individual throw, acting at a point midway between the two throws. The resultant R_1 of the centrifugal forces F_1 acting on the two inner throws is exactly equal to the resultant of the centrifugal forces acting on the two outer throws, and acts at the same point but in the opposite direction to the latter; consequently it neutralizes or balances it.

Local Balance—But while a crankshaft of this type is perfectly balanced as a whole, its individual throws are unbalanced, and since the crankshaft is more or less flexible, the centrifugal force acting on the individual throw tends to produce a pressure of the crank journals adjacent to it on their bearings.

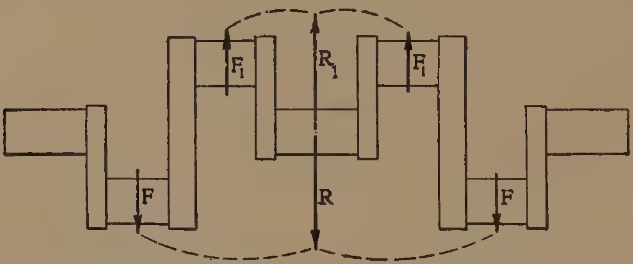


Fig. 122.—Centrifugal Forces on Four-Throw Crankshaft.

To eliminate this bearing load it is now customary to provide such arms of high-speed engines with balance weights. A rather serious degree of unbalance occurs in a four-cylinder, two-bearing crankshaft, because in the conventional design there is nothing to balance the two inner crankpins and the intervening portion of the crankshaft. Balance weights are now welded to crankshafts of this type, as well as to others, with the oxy-acetylene torch by means of a process controlled by the Park Drop Forge Co. The Wyman-Gordon Co. has developed a process of forging balance weights integral with multi-throw crankshafts.

Two counterbalanced four-cylinder crankshafts with the balance weights formed integral or welded on are shown in Fig. 123. The first of these is a three-bearing crankshaft and has extensions on all of its four short crank arms. The second is a two-bearing crankshaft, and in addition to balance weights on its two short crank arms, has a heavy balance weight welded to that portion of it between the two inner crankpins.

In a six-throw crankshaft the throws are arranged in pairs, the two inner ones being in line with each other, as are also the two outer ones and the two intermediate ones, respectively. Each pair of throws is located at an angular distance of 120° from the other two pairs. By reference to Fig. 124, which is an end view of a six-throw, seven-bearing crankshaft, it can easily be seen that if all of the throws are identical, such a crankshaft is in perfect balance as a whole. The resultant *F* of the centrifugal forces on the two throws of each pair acts at the middle of the length of the crankshaft. The three resultants therefore all act in the same plane radially outward from the center of the crankshaft, at angles of 120°, and they exactly balance each other. The highest degree of balance, of course, is obtained if each throw is balanced separately, which involves the use of a balance weight on each crank arm.

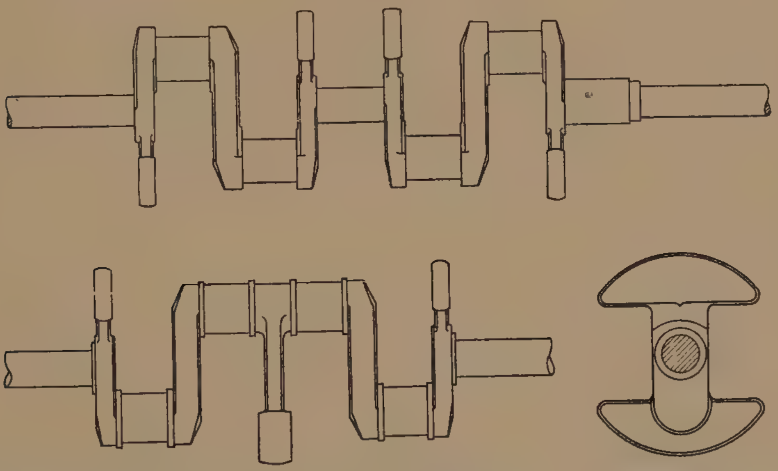
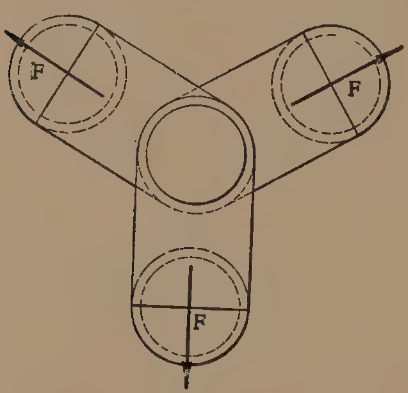


Fig. 123.—Four-Cylinder Crankshafts with Integral Balance Weights.

Six-cylinder crankshafts with either three or four main bearings are not inherently balanced. In fact, as sometimes made, they are not even in static balance. In the most primitive form, the long crank arms extend straight across from one crankpin to another at an angular distance of 120° therefrom. This crank arm then lies wholly to one side of the axis of rotation, and in operation produces an unbalanced rotating force whose direction is at all times along a line through the axis of rotation and the center of gravity of the crank arm. The magnitude of this unbalanced force is proportional to the distance of the center of gravity of the crank arm from the axis of rotation, and therefore can be reduced by reducing this distance, by curving the crank arm inward, as shown in Fig. 125. The rotating force, of course, can be entirely eliminated by bolting a balance weight against the side of the long crank arm, as shown in Fig. 126, which practice has been followed for many years by the Hudson Motor Car Co.

Crankshaft Proportions—An analysis of the stresses in the members of multiple-throw crankshafts involves very complicated mathematical reasoning and is of little practical value, as the stresses in normal operation, even under full throttle, are far below those which can cause failure. Breakage of crankshafts is now of very rare occurrence and seems to be possible only—barring defective material—if the engine is operated for a prolonged period at a critical speed. The elements of crankshafts, therefore, are calculated entirely by means of empirical formulae. Torsional stiffness is an important consideration in multiple-throw crankshafts, and the degree of rigidity required depends primarily on the inertia forces.



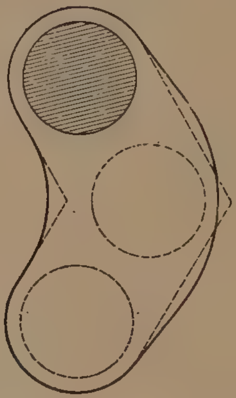


FIG. 124 (LEFT).—END VIEW OF SIX-THROW, SEVEN-BEARING CRANKSHAFT.

FIG. 125 (RIGHT).—CURVED LONG ARM OF SIX-THROW, THREE OR FOUR-BEARING CRANKSHAFT.

Except in the smaller sizes, four-cylinder engines are now generally provided with three bearing crankshafts. For passenger car work the four-cylinder engine is not very popular in the United States. There were only seventeen models with such engines on the American market at the beginning of 1925, and the number was decreasing. In

Europe, on the other hand, the majority of all passenger car models have four cylinder engines, and practically all trucks bulit both here and abroad are equipped with them.

The smallest passenger car engines (in Europe they range from about 2-in. bore up) are often designed with only two main bearings. This construction makes for cheapness of manufacture and for compactness. If an intermediate bearing is wanted it must be of substantial size, because the load on this bearing due to inertia forces is twice as great as the corresponding loads on the end bearings, which is due to the fact that the two sets of reciprocating masses on opposite sides of it are in phase and their inertia forces add together, while the end bearings have a set of reciprocating masses on one side of them only.

Three main bearings are used in the great majority of all four-cylinder engines, and the practice is being extended to larger engines, although a few of the heaviest types still have five main bearings. Four main bearings have been used in truck engines to a limited extent. The idea evidently was to eliminate the center bearing, which is subject to the heaviest loads, and to divide these loads among two bearings, between crank throws 1 and 2 and 3 and 4, respectively. Objections to the five-bearing crankshaft are that the space requirements for the bearings add to the over-all length of the engine and that in order to keep down the over-all length, the crank arms must be faced off on both sides, which adds to the expense. In the following are given empirical rules for dimensioning crankshafts of automobile engines based on a mass of carefully collected data of recent designs.

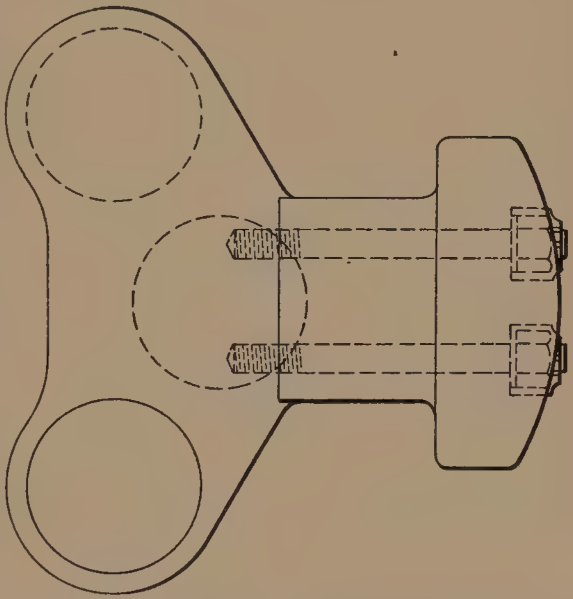


FIG. 126.—LONG CRANK ARM WITH BALANCE WEIGHT.

Four-Cylinder, Three-Bearing Crankshafts—In a great many cases the diameter of the crankpins and of all main bearings are made the same. If there is any difference, the former are the smaller, usually by 1/8 in. In many engines, however, the main bearings are stepped, decreasing by either 1/16 or 1/8 in. from the rear one forward. The reason for this is entirely one of ease of manufacture. It permits of boring out all of the main bearing seats simultaneously, with a multiple-cutter tool, inserted from the rear end. If it is desired to adopt this plan we may regard the diameter of the center bearing as the basic diameter and make the front and rear bearings smaller and larger in diameter, respectively, by either 1/16 or 1/8 in.

Denoting the cylinder bore by b; the crankpin diameter by d; the crankpin bearing length by l; the main bearing basic diameter by d_1; the front main bearing length by l_1; the center main bearing length by l_2 and the rear main bearing length by l_3, the average proportions of these dimensions in truck engine practice are as follows:

$$d = b/1.75; \qquad d_1 = b/1.75; \qquad l = b/1.65$$
$$l_1 = b/1.5; \qquad l_2 = b/1.35; \qquad l_3 = b/1.20$$

The above figures represent average present day practice, but the author believes that it is a better plan to make the center and rear bearings of the same length, about $b/1.28$. The practice of making the rear main bearing of extra length is a heritage from the early days of single-cylinder, low-speed, stationary engines, when large flywheels had to be used, the weights of which constituted a considerable bearing load. The practice proposed is more in accordance with the ratio of the loads on the center and rear bearings respectively, and, besides, if the bearings are not stepped, the two bushings will be identical, which is an advantage from both the manufacturing and the servicing standpoints.

An analysis of the crankshaft dimensions of eight four-cylinder, small-bore, three-bearing passenger car engines (foreign) gave the following average relations:

$$d \quad b/1.72 \qquad d_1 \quad b/1.65 \qquad l \quad b/1.66$$
$$l_1 \quad b/1.34 \qquad l_2 \quad b/1.55 \qquad l_3 \quad b/1.11$$

Several of these small foreign engines have counterbalanced crankshafts, which permit of the use of somewhat shorter main bearings.

It will be noted that these small high speed engine crankshafts have slightly greater journal diameters, relatively, than the truck engine crankshafts. The front and rear bearings are rather longer than in the truck engines, which is due to a large extent to exigencies of design, while the center bearing is shorter, no doubt because of the greater importance of limiting bulk and weight in the case of the passenger car engine.

Crank Arms—The relative width of the crank arms varies with the number of throws between supporting bearings. For instance, in a crankshaft with four throws between supports, the bending stress in the plane of the crankshaft will be very great and the material in the crank arms, therefore, must be so disposed as to give the maximum resistance to bending stresses in this plane; that is, the arms must be made thick and rather narrow. On the other hand, if there is only a single throw between bearings, space in the direction of the crankshaft axis is usually limited, in view of which the thickness of the arms is reduced and the necessary resistance to bending of the arms is secured by adding to their width.

Another consideration that has an influence on the width of the crank arms is the necessity of providing means for taking up thrust due to outside causes, such as a helical gear camshaft drive at the forward end or pull or push on the clutch collar when the clutch is disengaged, at the rear end. If the crank arms are made wide they naturally provide shoulders at the crank journals on which this thrust load may be taken up. In the case of crankshafts with narrow crank arms, circular flanges are sometimes forged on to both sides of the center journal, as shown in Fig. 129, or at both sides of the rear journal, as exemplified in Fig. 128. In such cases the width of the crank arm can be proportioned entirely in accordance with requirements of stiffness.

The thickness of the short crank arm of course, is made less than that of the long arm, because the bending moment on the former is smaller. The resistance to bending of the arms in the plane of the crankshaft should be proportional to the resistance to bending of the crankpins, that of the short crank arms being made somewhat smaller and that of the long crank arms (in a four-cylinder three-bearing crankshaft) somewhat greater. The following proportions give good results:

$$w = 0.7b \qquad t_s = 0.28b \qquad t_l = 0.38b$$

In all drop-forged, unfinished crankshafts, the arm section, of course, is not a rectangle, as draft has to be allowed on the two long sides, usually about 7°. In applying

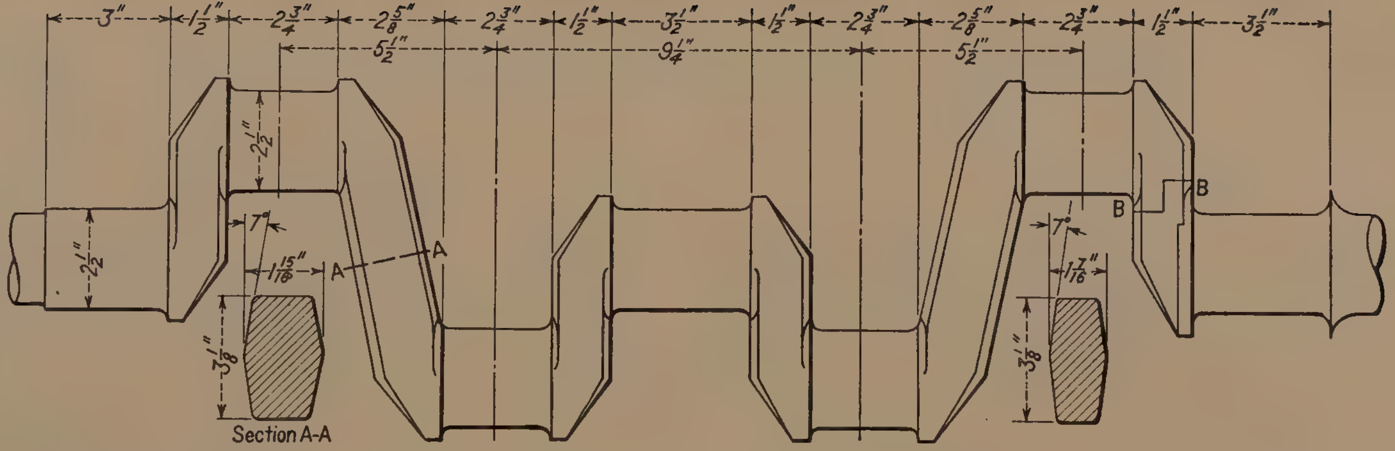


FIG. 127. — THREE-BEARING CRANKSHAFT FOR FOUR-CYLINDER 4½ BY 6-IN. TRUCK ENGINE.

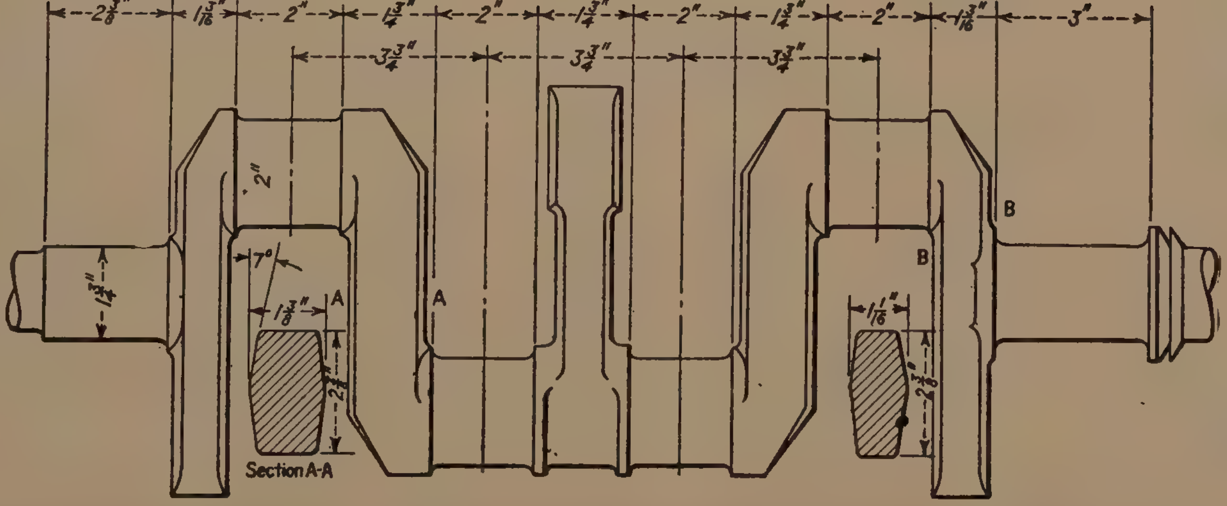


FIG. 128. — TWO-BEARING CRANKSHAFT FOR FOUR-CYLINDER 3 BY 4½-IN. ENGINE.

the above formulae for thickness, the calculated thickness can be taken as the mean between the minimum and maximum actual thicknesses.

Applying the equations to a crankshaft for a four-cylinder 4½ by 6-in. truck engine, we obtain the following results:

$$d = 4.5/1.75 = 2.57, \text{ say } 2\tfrac{1}{2} \text{ in.}$$
$$l = 4.5/1.65 = 2.78, \text{ say } 2\tfrac{3}{4} \text{ in.}$$
$$d_1 = 4.5/1.75 = 2.57, \text{ say } 2\tfrac{1}{2} \text{ in.}$$
$$l_1 = 4.5/1.5 = 3.00, \quad 3.00 \text{ in.}$$
$$l_2 = 4.5/1.28 = 3.52, \text{ say } 3\tfrac{1}{2} \text{ in.}$$
$$l_3 = 4.5/1.28 = 3.52, \text{ say } 3\tfrac{1}{2} \text{ in.}$$
$$w = 0.7 \times 4.5 = 3.15, \text{ say } 3\tfrac{1}{8} \text{ in.}$$
$$t_s = 0.28 \times 4.5 = 1.26, \text{ say } 1\tfrac{1}{4} \text{ in.}$$
$$t_1 = 0.38 \times 4.5 = 1.71, \text{ say } 1\tfrac{3}{4} \text{ in.}$$

Four-Cylinder, Two-bearing Crankshaft—This type, as a rule, is used only in engines of less than 200 cu. in. piston displacement. Contrary to practice with other types, the crankpins sometimes are made of larger diameter than the main journals, which does not seem to be illogical in view of the variation of the bending moment. The following rules apply:

$$d = b/1.55 \qquad l = b/1.55 \qquad d_1 = b/1.70$$
$$l_1 = b/1.25 \qquad l_2 = b \qquad w = b/1.25$$
$$t_s = 0.3\,b \qquad t_1 = 0.4\,b$$

These proportions are based on the assumption that balance weights are provided both on the short crank arms and at the middle of the crankshaft. The two-bearing type of crankshaft is often used where the shaft is to be fitted with ball bearings. Such crankshafts will be discussed later on.

In a crankshaft of this type if the thrust to be provided for comes from the rear, it is well to take up end thrust in both directions on thrust collars to both sides of the rear main journal, as shown in the figure. This entirely relieves the cranks of end thrust resulting from forces outside the engine. When provision is thus made to take up the thrust at certain points, a slight clearance must be allowed at other points where thrust might be taken up, as on the inside of the front main journal in this case.

Six-Cylinder Three-Bearing Crankshafts—Crankshafts of six-cylinder engines are supported on three, four or seven bearings, the four-bearing construction being the most popular. Of the 1925 American six-cylinder models, 27 per cent had three-bearing crankshafts, 53 per cent four-bearing

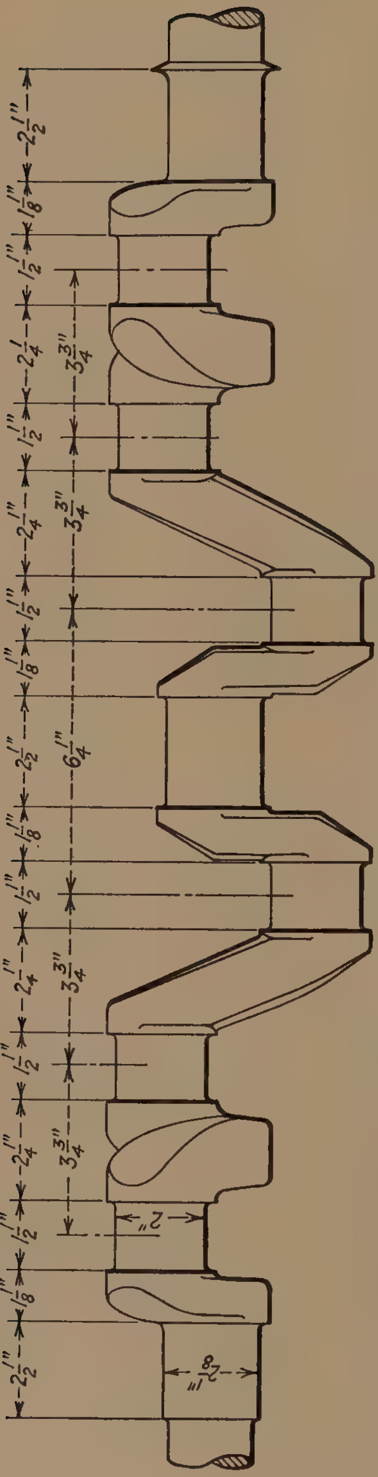


FIG. 129.—THREE-BEARING CRANKSHAFT FOR SIX-CYLINDER 3 BY 4½-IN. ENGINE.

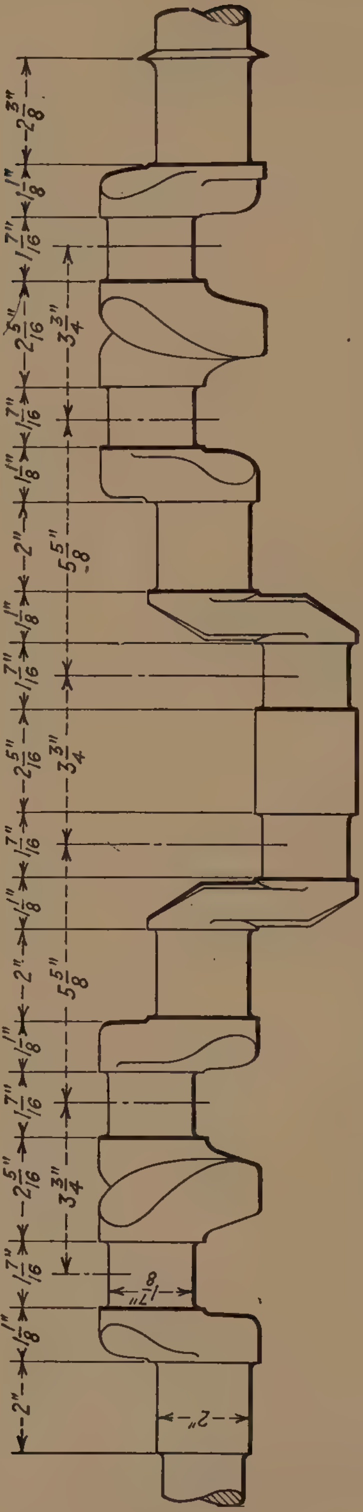


FIG. 130.—FOUR-BEARING CRANKSHAFT FOR SIX-CYLINDER 3 BY 4½-IN. ENGINE.

crankshafts and 20 per cent seven-bearing crankshafts. On the whole the tendency is to use seven-bearing cranks in the larger and higher grade engines and this type seems to be gaining ground at present. In the case of a three-bearing crankshaft, since there are three throws between supports, the crankpins and crank arms must be of very substantial size. Following are rules for the dimensions of crankshafts of this type:

$$\begin{array}{lll} d = b/1.50 & l = b/2 & d_1 = b/1.45 \\ l_1 = b/1.45 & l_2 = b/1.20 & l_3 = b/1.20 \\ w = b/1.28 & t_s = 0.28\,b & t_l = 0.5\,b \end{array}$$

The three-bearing crankshaft is used more particularly in smaller engines, but it has lost a great deal of ground since 1915 when it was used in nearly one-half of all six-cylinder passenger-car engines. One possible reason for its decline is that it does not lend itself so well to full force feed lubrication as the other types, requiring an outside tube on two of the long crank arms. In order to reduce the bearing loads due to the unbalanced long crank arms, these must be curved inward as much as possible, and balance weights forged on or applied would further reduce these loads. The above rules applied to a crankshaft for a 3 by 4 1/2 in. engine give the following dimensions: $d = 2$ in.; $l = 1\ 1/2$ in.; $d_1 = 2\ 1/16$ in.; $l_1 = 2\ 1/16$ in.; $l_2 = 2.5$ in.; $l_3 = 2.5$ in.; $w = 2\ 5/16$ in.; $t_s = 1\ 7/8$ in.; $t_l = 1.5$ in. A drawing of such a crankshaft is shown in Fig. 129.

Six-Cylinder Four-Bearing Crankshaft—Owing to the fact that there are only two throws between supports, the crankpins and arms can be made lighter, relatively, than in the previously described type. Following are rules for dimensions:

$$\begin{array}{llll} d = b/1.60 & l = b/2.05 & d_1 = b/1.5 & \\ l_1 = b/1.55 & l_2 = b/2 & l_3 = b/2 & \\ l_4 = b/1.25 & w = b/1.28 & t_s = 0.3\,b & t_l = 0.45\,b \end{array}$$

For a 3 by 4½ in. engine these rules give the dimensions shown on the crankshaft illustrated in Fig. 130.

Six-Cylinder Seven-Bearing Crankshafts—In a seven-bearing crankshaft the tendency to torsional vibration is probably greater than in any other type, owing to the greater aggregate length of the main journals. On the other hand, the bending moments are small, owing to the short distances between supports, hence it seems logical to make the crankpins of smaller diameter than the main journals. As an

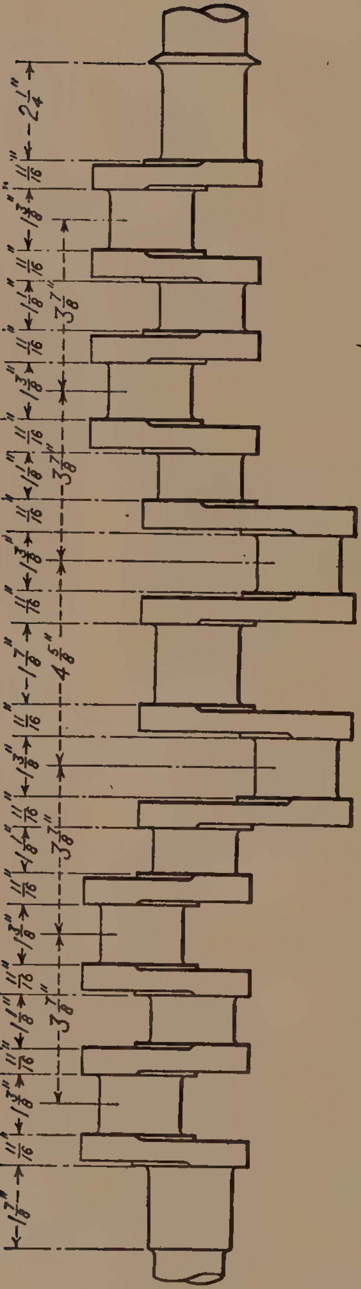


Fig. 131.—Seven-Bearing Crankshaft for Six-Cylinder 3 by 4½-In. Engine.

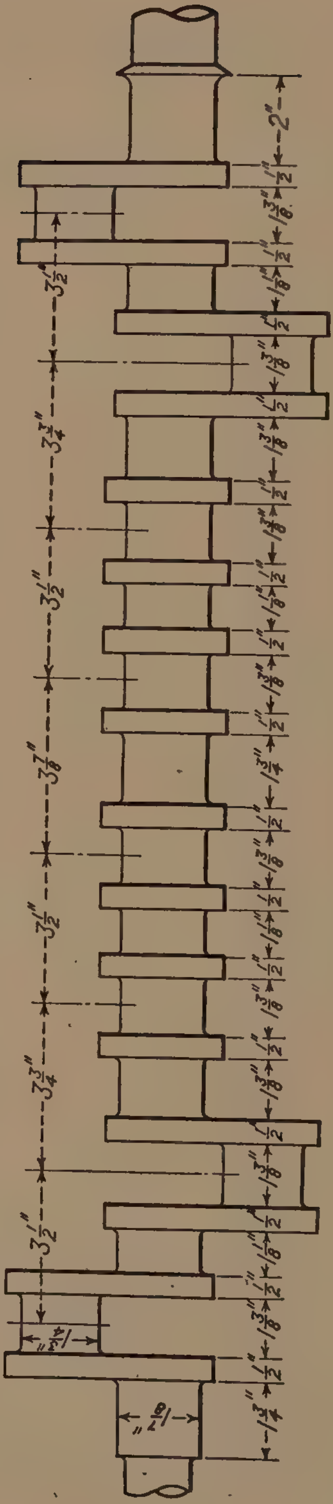


Fig. 132.—Nine-Bearing Crankshaft for Eight-Cylinder Vertical 2¾ by 4½-In. Engine.

instance of the extremes to which some designers have gone to prevent torsional vibration in six-cylinder engines it may be mentioned that the Sheffield Simplex six-cylinder 3½ × 5 inch engine has a seven-bearing crankshaft with main journals 2¾ inches in diameter. To secure adequate bearing surface without unnecessarily lengthening the engine the crank arms must be finished on the sides at least and the journals must come right up to the sides of the arms.

Rules for the dimensions of this type of crankshaft are as follows:

$$d = b/1.65 \qquad l = b/2.25 \qquad d_1 = b/1.45$$
$$l_1 = b/1.6 \qquad l_2, l_3, l_5, l_6 = b/2.75 \qquad l_4 = b/1.65$$
$$l_7 = b/1.35 \qquad w = b/1.10 \qquad t_s = 0.20\, b$$

These rules applied to a crankshaft for a six-cylinder 3 by 4½-in. engine give a crankshaft of the dimensions shown in Fig. 131.

Nine-Bearing Crankshaft for Eight-Cylinder Vertical Engine—To secure perfect balance of reciprocating parts the four throws at the middle must be in one plane and the remaining four throws in a plane at right angles thereto. Some crankshafts for eight-cylinder vertical engines have been made with two four-cylinder cranks in planes at right angles to each other joined end to end, but this produces a rocking couple in the vertical axial plane. The following proportions may be used:

$$d = b/1.60 \qquad l = b/2 \qquad d_1 = b/1.45 \qquad l_1 = b/1.55$$
$$l_2, l_4, l_6, l_8 = b/2.45 \qquad l_3, l_7 = b/2 \qquad l_5 = b/1.7$$
$$l_9 = 1.35 \qquad w = b \qquad t_s = 0.20\, b$$

A crankshaft for a vertical eight-cylinder 2¾ by 4½-in. engine is shown in Fig. 132.

Three-Bearing Crankshafts for Eight-Cylinder Engines—In an eight-cylinder engine with two sets of cylinders at an angle of 60° or 90°, any of the types of crankshaft suitable for a four-cylinder engine—two, three, four and five bearing—may be used, but the two- and three-bearing types are the most advantageous. In such engines two connecting rods connect to each crankpin, and there are three different methods of arranging the connecting rod big-end bearings, which affect the design of the crankshaft. These methods are as follows:

1. One connecting rod yoked and clamped to a bearing bushing, the other having a bearing on the outside of the bushing.

2. Bearings on crankpin side by side.

3. One "master" connecting rod, having its bearing on the crankpin, the other hinged to the master connecting rod.

The yoked form of construction is the most popular. In proportion to the cylinder displacement the crankpin area will be large, and this is a very desirable feature, as with the ordinary yoked construction the crankpin bearing cannot be refitted when worn, but must have its bushing replaced. For an eight-cylinder three-bearing crankshaft with yoked connecting rods the dimensions may be made as follows:

$d = b/1.35$ $l = b/1.10$ $d_1 = b/1.35$
$l_1 = b/1.15$ $l_2 = b/1.05$ $l_3 = b/0.85$
$w = b/1.08$ $t_s = 0.28\ b$ $t_l = 0.38\ b$

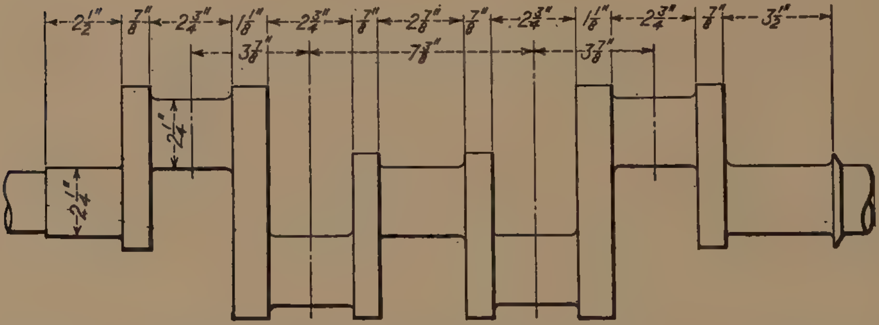


Fig. 133.—Three-Bearing Crankshaft for Eight-Cylinder 3 by 4½-In. V Engine.

A design of crankshaft in accordance with these rules, for an eight-cylinder V engine of 3-in. bore and 4½-in. stroke is shown in Fig. 133.

The second and third cylinders must be a considerable distance apart in order to make room for the middle bearing. To keep down the length of the engine it is well to face off the short crank arms on both sides and bring the journal right up to the finished surfaces. The long crank arms, however, can be made of the usual form with draft and left rough. Wherever the crank arms are finished as here proposed, a shoulder of 1/16 or 1/32 inch is provided at each end of each bearing, which facilitates the grinding operation.

Crank Journals Side by Side—If the crankpin journals are to be placed side by side (in which case the two cylinder blocks must be offset) the proportions of the crankshaft can be made the same as above, with the exception of the lengths of the crankpin bearings, which are made as long as the distance between adjacent cylinders permits, the long crank arms also being finished on the sides. For instance, in the above case the distance between axes of adjacent cylinders would be about 3 3/4 inches, and as the long crank arm occupies 1 1/8 inches of this space there remains 2 5/8 inches for the crankpin, making the bearing surface of each connecting rod 2 1/4 × 1 5/16 inches.

While at one time quite a few eight-cylinder V engines with two-bearing crankshafts were built, there are none being made now, and it does not seem necessary, therefore, to give rules for proportioning such crankshafts.

Fillets—At all changes in section there must be no sharp corners, as this tends to localize stresses and induce failure. Liberal fillets should be provided at the joints of main journals and crankpins with crank arms.

Front and Rear Ends—The front end of the crankshaft must be extended to carry the driving pinion of the camshaft gear and the pin or ratchet toothed sleeve with which the starting crank engages. There may also be an oil guard formed on the shaft inside the seat for the cam gear pinion. The starting ratchet seat is made smaller in diameter than the pinion seat, and the latter smaller than the crank journal.

At the rear end of the crankshaft an oil guard is usually provided, especially if a dry clutch is to be employed. This consists of a narrow flange turned on the crankshaft just beyond the rear crank journal, with a sharp outer edge, from which the oil is flung off when the shaft turns rapidly. The oil is caught in a recess turned in the outer end of the bearing hub, from which it drains back to the crank case.

Beyond the oil guard there is generally provided an integral flange to which the flywheel is bolted. This flange is made of an outside diameter about equal to the stroke, of a width equal to 1/16 inch per inch of bore multiplied by the square root of the number of cylinders and is drilled with six or more holes for the reception of bolts. The diameter of the bolts may be made equal to the width of the flange. Where a very secure job is desired the holes in the flange and in the web of the flywheel are counterbored and have hardened dowel bushings inserted, which take up the driving strain.

The above empirical rules for the dimensions of the flange give workable results, but a better plan is to determine the maximum shearing stress on the bolts from the maximum torque and allow a safety factor of 6.

In American practice the flywheel flange usually forms the rearmost part of the crankshaft, the crank being drilled out to a certain depth from the rear end to form a pilot bearing for the clutch shaft. In most European designs the crankshaft extends some distance beyond the flywheel flange and forms a pilot for the clutch.

Ball-bearing Crankshafts—Ball bearings are used to a certain extent for supporting engine crankshafts, but not in American passenger car practice. Three American builders of truck and tractor engines use them, and abroad they are used to a limited extent for both passenger-car and commercial-vehicle engines. The advantages are reduced friction, particularly while starting the engine in cold weather, and hence reduced wear and tear on the battery; in two-bearing, four-throw crankshafts no binding due to crankshaft flexure; reduction in overall length of engine and reduced maintenance cost.

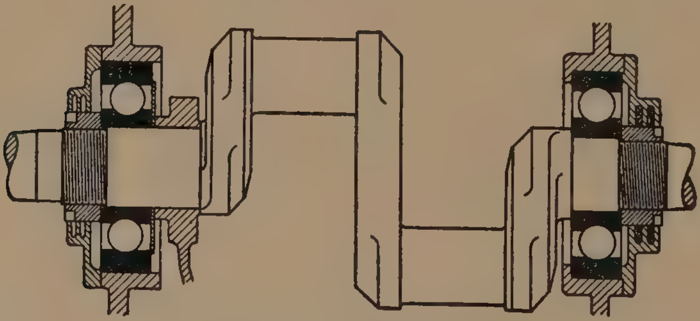
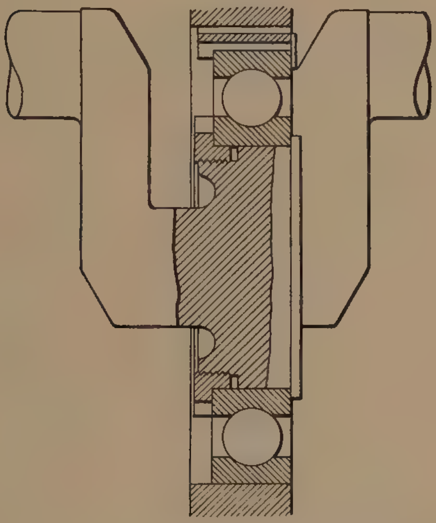


FIG. 134.—BALL-BEARING CRANKSHAFT OF TRUCK ENGINE (AUTOCAR).

These advantages are offset by greater first cost, less silent operation and difficulty of mounting if intermediate bearings are required. Ball-bearing crankshafts were introduced in American passenger-car practice about 1906, but they were soon given up, because of difficulties from noisy operation.

Four-cylinder crankshafts supported by only two ball bearings are used in quite large commercial vehicle engines, in the case of which silence is a minor factor. The flywheel cannot be secured to an integral flange but must be keyed to the shaft. The ball-bearing crankshaft of the Autocar truck engine is shown in Fig. 134.

Intermediate Ball Bearings—Two methods of mounting the intermediate bearing of a three-bearing crankshaft are illustrated in Figs. 135 and 136. The design in which the bearing is stripped over the crank arm is not very attractive in appearance, but since it is hidden in the crankcase this is of minor importance. The ends of the crank arms must be rounded to conform to the bore of the inner race. The intermediate journal must be of slightly more than twice the width of the ball bearing, and either an integral collar or a separate split collar and a spacer must be provided. There are various methods of forming the joints of built-up crankshafts. One consists in forming half of the main journal on each section, each of the halves having jaws and the two pairs of jaws engaging each other. The parts are then drawn together by a bolt passed through the hollow journal.



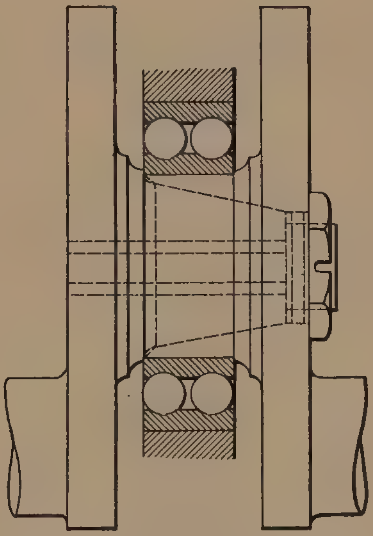


Fig. 135 (left).—Intermediate Ball Bearing Stripped over Crank Arm.

Fig. 136 (right).—Joint of Built-Up, Ball-Bearing Crankshaft.

An idea of the sizes of ball bearings required on crankshafts for engines of different displacements may be obtained from Table VI on page 236.

Drilled-out Crankshafts—In some of the higher grade engines the crank journals and crankpins are drilled out to a considerable diameter, to reduce their weight, the material near the axis of the journals adding little to their strength either in bending or in torsion. A typical design of this type is the Lincoln, illustrated in Fig. 137. In this case

Table VI—Crankshaft Ball Bearings

Name of Engine	Bore and Stroke (Inches)	Displacement (Cu. in.)	Front Bearing	Rear Bearing
Dorman	2.56 × 3.94	81	No. 409*	No. 410*
Bristol	2.56 × 3.94	81	No. 406*	No. 212*
Astral	2.72 × 4.53	105	No. 409*	No. 409*
White	3.75 × 5.12	226	No. 410	No. 412
A. E. C.	3.94 × 5.52	268	No. 315	No. 316
International	4.25 × 5.00	283	No. 314	No. 316
Autocar	4.25 × 5.50	312	No. 413	No. 316
Comercar†	4.72 × 5.50	385	No. 312	No. 221
Saurer‡	4.33 × 7.10	417	No. 411	No. 221

* Wide Series; † Center bearing, No. 219; ‡ Center bearing, No. 220, All of the engines referred to in the above table are four-cylinder engines. the first seven having two main bearings and the last two, three.

1-inch holes are drilled through both the main journals and the crank arms. The holes through the five main journals are drilled in one operation with a drill almost 3 ft. long, in an upright drilling machine, the work being held in a special two-station, indexing jig in which a crankshaft can be reloaded while another is being drilled. The holes through the four crankpin bearings are drilled in a somewhat similar set-up, a double-spindle drill head being used to drill simultaneously the two crankpins on opposite sides of the crankshaft. Radial holes of smaller diameter are drilled to connect the holes in the crank journals and crankpins, respectively, and the open ends of all holes are plugged. Later the tightness of these screw plug closures is tested with kerosene under a pressure of 75 lbs. per sq. in. The holes in the crankpins are first drilled to 15/16 inch and they are later bored out to 1 inch, to correct any "run-out" of the long drills and eliminate its effect on the balance. The passage through the crankshaft is used for oil feed to the bearing surfaces and a small diameter radial hole is drilled in each journal, in that particular angular position in which the load on the bearing is a minimum.

Inclined Oil Holes—The use of radial oil holes in the crankpins is somewhat objectionable, as the hole comes at that part of the pin which is subjected to the maximum bending stress, and materially reduces its strength. This objection is overcome by drilling diagonal holes through the crank arms from the journal to the crankpin, as illustrated in Fig. 138.

When the oil is to be fed to crankpin bearings under pressure through holes drilled in the crankshaft, difficulty is encountered in the case of a crankpin with "long" crankarms on both sides of it. If an oil hole were drilled through such an arm the oil would have to pass through it, part way, in a direction toward the axis of rotation, against centrifugal force. The oil would previously have passed by one main and one crankpin bearing, and if these were at all loose it would pass out through them freely and there would not be sufficient pressure left to overcome the centrifugal force. For this reason, in a case of this kind, oil holes are drilled into the arm from the sides approximately at the axes of the crankpins, and a tube is secured into this hole, of such shape that every portion of it is at substantially the same distance from the axis of rotation (Fig. 139). This plan is used with four-cylinder two-bearing and six-cylinder three-bearing crankshafts of engines with pressure lubrication.

Crankshaft Forgings—The forgings from which crankshafts are machined are made in dies under drop hammers in a number of operations. The stock comes in the form of either square or round bars and is sheared off to length. During the forging process the scale formed is removed by means of a jet of compressed air after each operation, and after the forging process has been completed the remaining scale is removed by pickling.

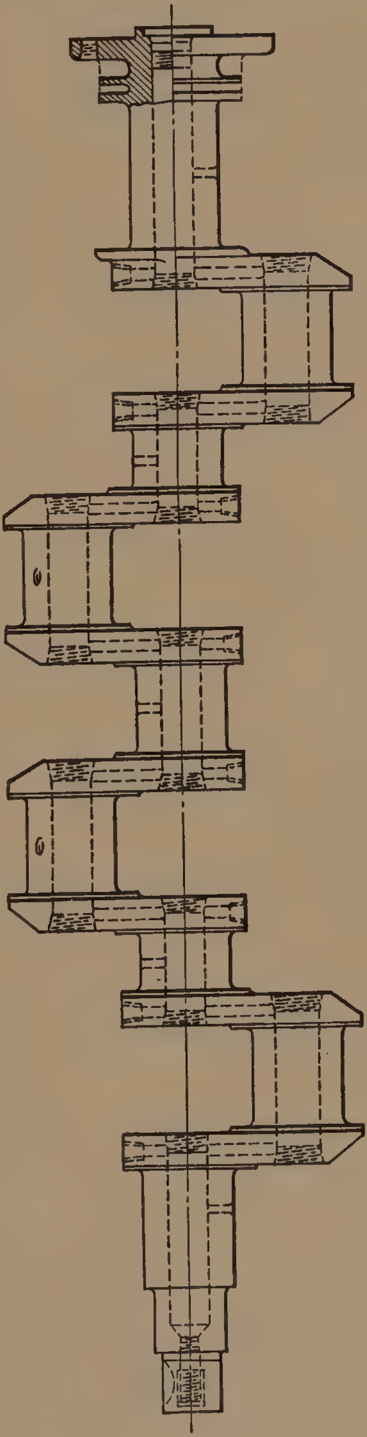


FIG. 137.—DRILLED-OUT CRANKSHAFT (LINCOLN).

With the increase in the rotative speed of engines the need for balance weights on crank arms has become more urgent. Bolting the balance weights to the arms is more or less unsatisfactory, owing to the machine work involved and the great strains on the bolts at high speeds. Drop forgers have sought to meet the situation by providing integral balance weights on their crank forgings. According to Wyman & Gordon, although it is possible to forge on balance weights of any size wanted, a compromise is usually made between complete balance and economical production.

In order to be able to readily withdraw the forging from the die, a draft must be allowed in the design of the forging, which is generally set down at 7°. The need for the draft adds to the difficulty of providing adequate balance weights, as the thickness of the forging must decrease from the axis outward, whereas the greatest balancing effect would be obtained with the least material if the thickness increased from the axis outward. In endeavoring to minimize this handicap it has been found possible to reduce the draft on the balance weights to 3°.

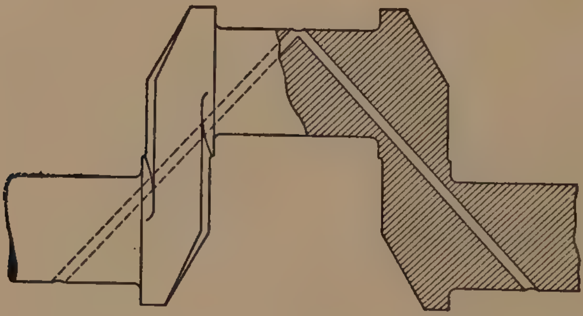


FIG. 138.—DIAGONALLY DRILLED CRANKSHAFT.

Crankshafts with throws in more than one plane are forged in the first place with all throws in the same plane and are then twisted or indexed, while hot, in a special machine. Accurate indexing (which means that the angles between axial planes of the several throws must be equal) and accurate alignment of the main journals are very important requirements. Moreover, the crankshaft must be accurately centered, because many shafts are not machined on the crank arms and would be likely to be far out of balance if inaccurately centered. First the crank is adjusted in a mass centering machine to give the best possible index, and it is then centered on the axis determined by this operation. Accurate centering is even more important in the case of "single-plane" (four-cylinder) than in that of multiplane (six-cylinder) crankshafts, as if the center is materially off the mass axis in the former the unbalance is very great.

Machining Crankshafts—Following are some notes on major operations in the machining of a four-throw crankshaft, those on a six-throw being similar except that there are one more pair of crankpins to be finished.

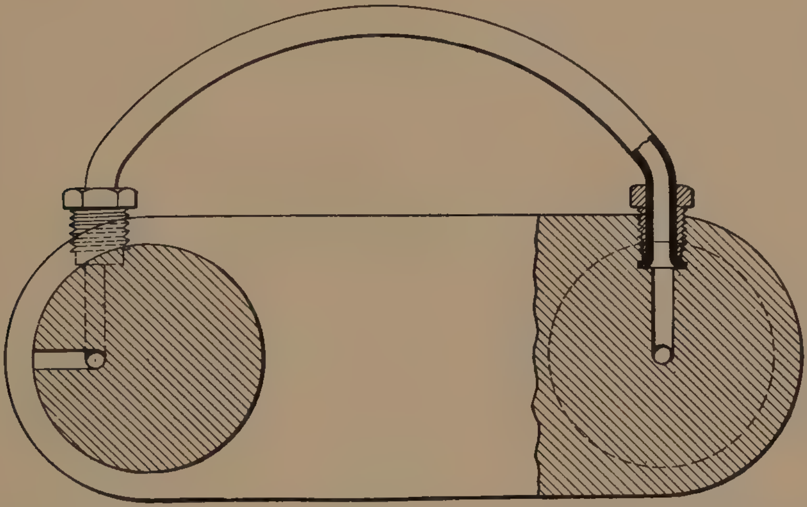


FIG. 139.—OIL TUBE CONNECTING OPPOSITE ENDS OF LONG CRANK ARM.

After the alignment and indexing of the crank have been checked in a fixture—the shaft having been centered previously—the center bearing is first rough-turned, for which operation the crankshaft is held between centers and steadied by a potchuck gripping it near the center. An automatic compensating chuck for gripping and supporting the crankshaft by one of the inner throws has been worked out by the Jones & Lamson Machine Co. It is supported by a large steady-rest. Both the center main bearing and the inner faces of the two adjacent arms are finished in one operation. A special spring center is used in the headstock, and a gage block serves to locate the crankshaft endwise, so as to divide the stock equally between the throws adjacent to the center bearing.

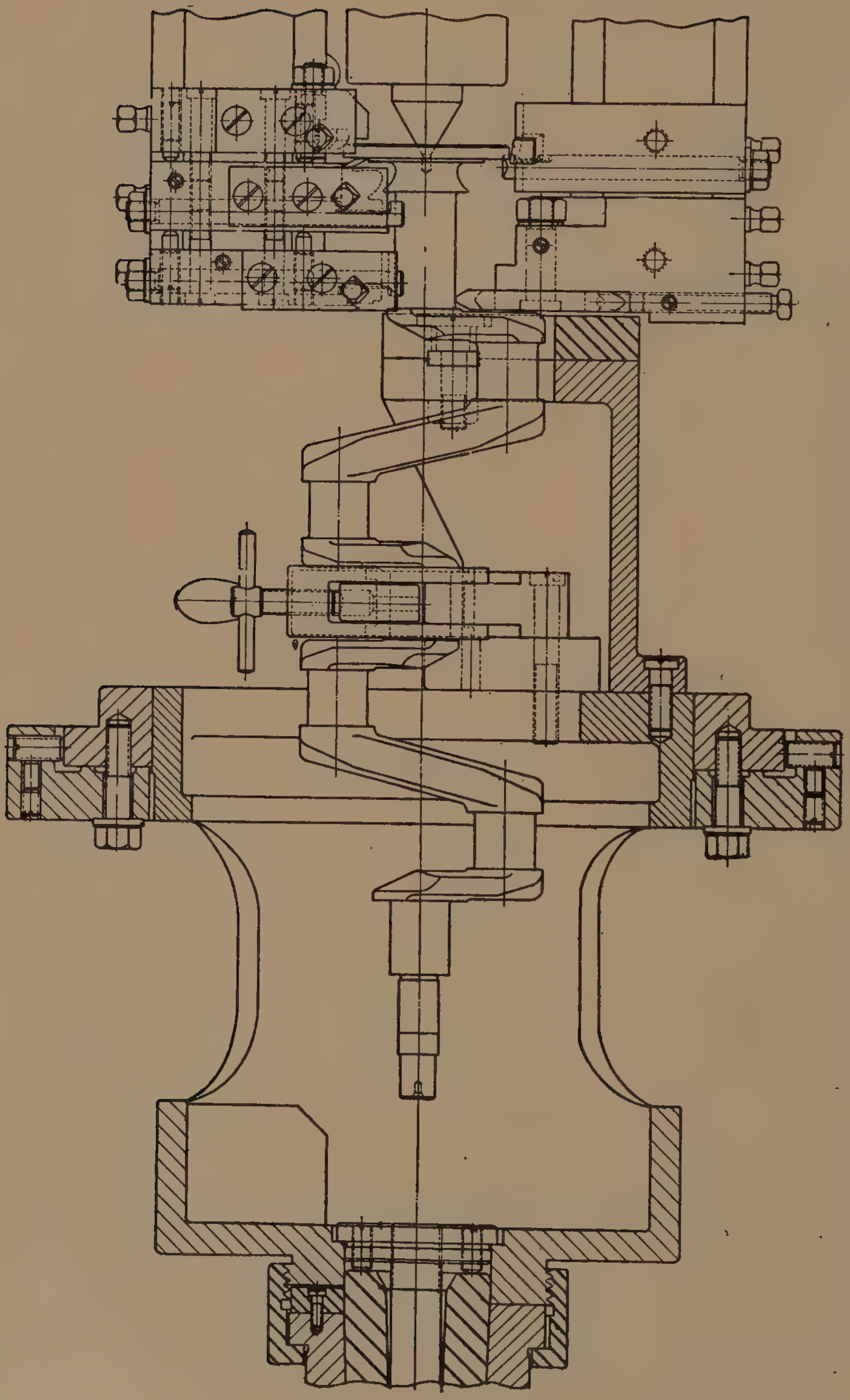


Fig. 140.—Tool Lay-out for Turning and Facing Flywheel End of Crankshaft in Fay Lathe.

In the next operation the forward or gear end of the crankshaft is turned. The shaft is held in a large potchuck with hinged locating and gripping plate gripping the center bearing and the tailstock center. A number of tools in two tool carriages are at work simultaneously. Next similar operations are performed on the driving end, where the flywheel flange also has to be turned and faced. The method of holding the crankshaft in the Fay automatic lathe and the tooling for this operation are illustrated in Fig. 140, while the lathe tooled up for this operation, with the crankshaft in place, is shown in Fig. 141.

Turning Crankpins—Crankpins are generally turned on special lathes which are installed in sets equal to the number of pairs of pins to be turned. For instance, for a six-throw crankshaft, three crankpin turning lathes are used. With the Wickes Duplex crankpin turning lathe there are two tools at the front with which the center portion of the pin is turned, after which the rear tools are brought into action, which turn the end portions and fillets.

The crankshafts come to these lathes after they have been roughturned on the front, rear and center main journals. They are clamped in cast steel potchucks mounted on the flanged spindles of both headstocks. Finishing of the pin is completed in less than a minute, the machine being operated semi-automatically. The crankpin turning operation is illustrated in Fig. 142.

Form-grinding Crankarms—During the War, when aircraft engines were in production in a good many automotive plants, the practice of finishing the crankshafts all over became familiar, and some manufacturers later introduced it in connection with automobile engines. Six-cylinder cranks with seven main bearings and eight-cylinder with nine bearings require the crank arms to be faced on both sides in any case, in order to minimize the overall length, and if that much must be finished, there is, of course, an incentive to finish all surfaces, which gives a crankshaft that is very nearly balanced when completed.

Special machines have been developed for turning the crank arms to an elliptical or similar form, and one such machine, the Gordon Form lathe, installed in the Packard shops, is illustrated in Fig. 143. The crankshaft is supported in bearings or steady rests at all intermediate bearings and all of the arms are turned at once, the tool slides being moved in and out by "masters" the same as in cam turning or grinding. In some cases the arm contours are ground to form.

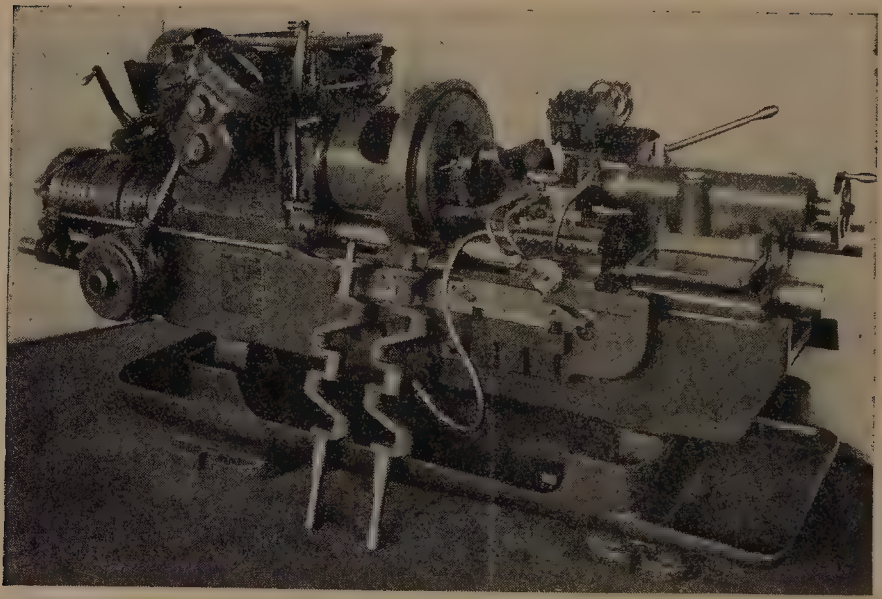


FIG. 141.—TURNING AND FACING FLYWHEEL END OF CRANKSHAFT IN FAY LATHE.

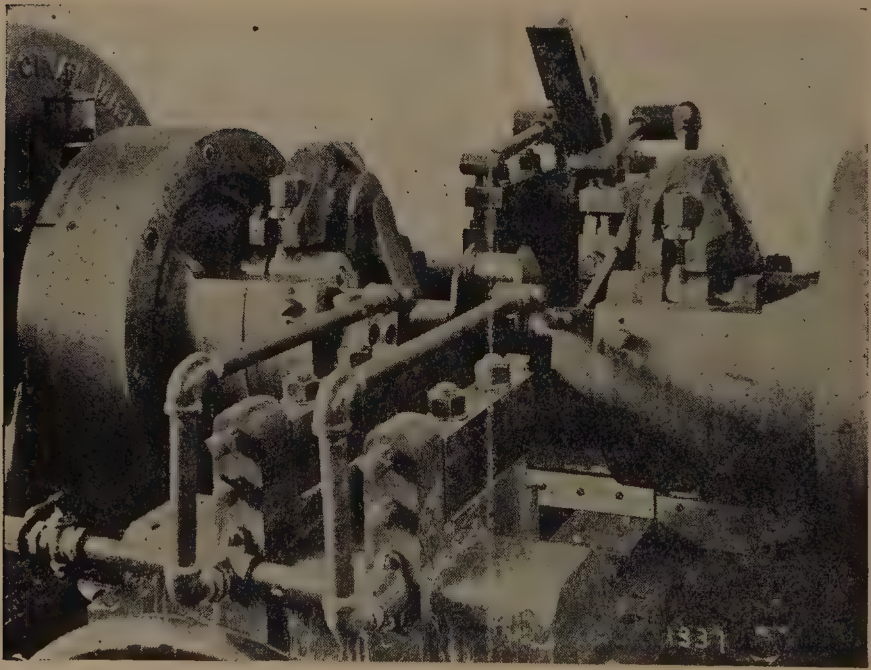


FIG. 142.—TURNING CRANKPINS IN WICKES BROTHERS CRANKPIN LATHE.

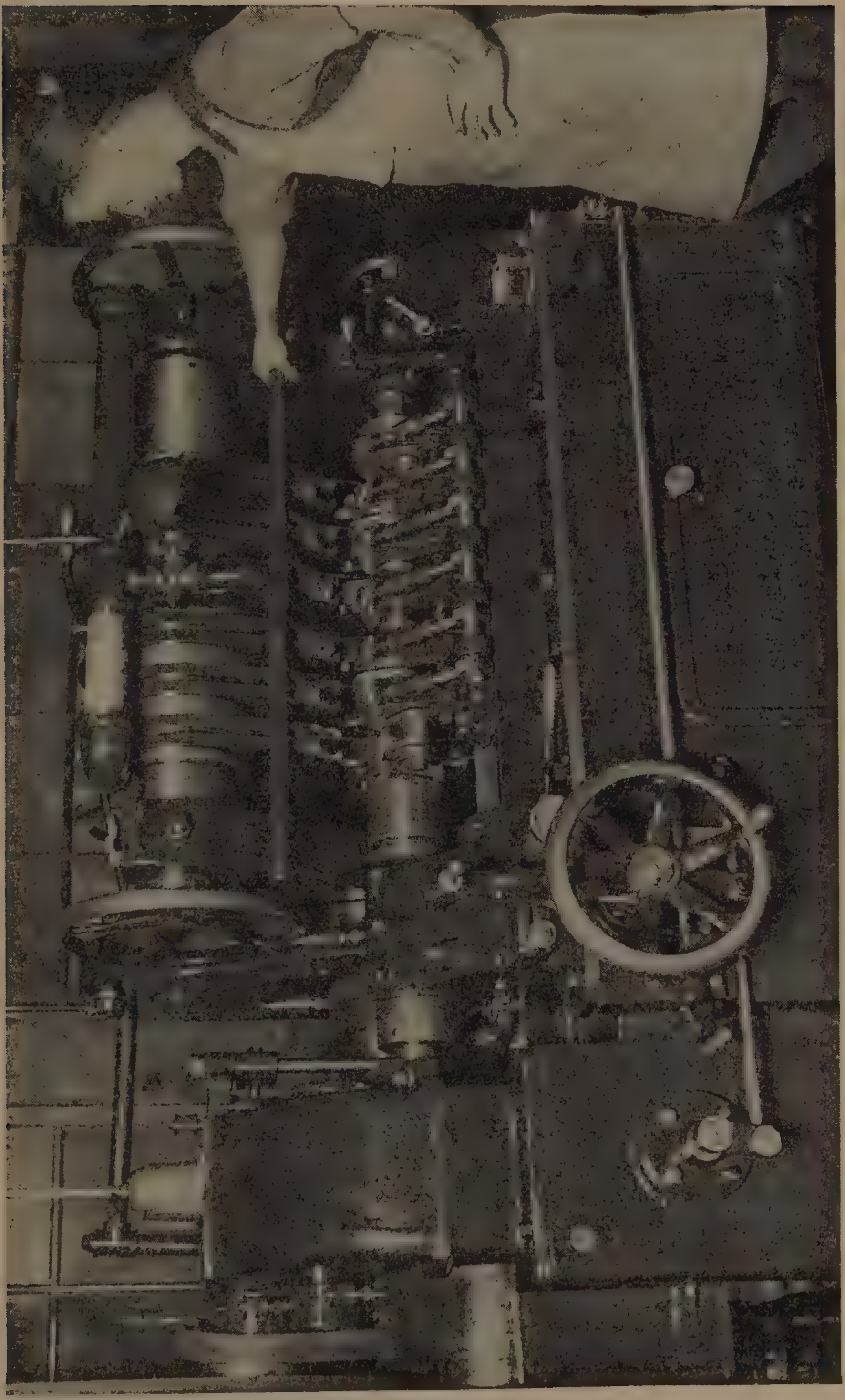


FIG. 143.—TURNING PACKARD CRANK ARMS IN GORDON FORM LATHE.

Sometimes the crankshafts are normalized between the roughing and finishing operations, to remove any stresses set up by the machining. In one instance this is effected by submerging the crankshafts, held in a large wire mesh basket, in an electrically heated sodium nitrate bath at 850° F. for 45 minutes.

Drilling Oil Holes—Drilling oil holes through the arms of crankshafts is a rather difficult operation. Inasmuch as the oil hole weakens the shaft to a certain extent, it is generally made of quite small diameter, and as the hole is quite deep, the strain on the drill is great. If hand feed is used, the drills are often broken, which necessitates a very onerous salvage operation.

A method which has practically eliminated drill breakages has been developed in the shops of the H. H. Franklin Mfg. Co. The six-throw crankshaft is held at the required angle in a fixture, and two holes are drilled at the same time. A notched cam takes the place of the usual hand lever for controlling the feed of the drills. The cam is similar in appearance to a coarse-tooth circular saw. There are spiral surfaces corresponding to the clearance surface of the saw teeth, and when the roller-ended feed lever of the drill contacts with such a surface the drill heads are fed forward at a constant rate. As the roller drops over what corresponds to the cutting edge of the saw tooth, a heavy counterpoise lifts the drills out of the holes and thus frees them of chips. There are nine "teeth" on the cam, and as the holes are drilled halfway through the arm from each end, this ensures very effective clearing of chips.

Grinding Crankshafts—All main journals and crankpins are finished by grinding, sometimes in two operations. In the rough-grinding process the journals are ground to within 0.003 or 0.005 inch of finish size. The grinding of the pins is usually accomplished by means of a straight-in feed. This means that the grinding wheel used must be of the same width as the distance between shoulders, and this width has been standardized by the S. A. E.

In grinding crankshafts, the cost of the diamonds for dressing the grinding wheel and the wages of the wheel-dresser are important items of expense, and all fillets on the crankshaft therefore should be made of the same radius, so that the amount of dressing required may be reduced to a minimum.

After the crankshaft is completed it is carefully inspected for accuracy of diameters, roundness, indexing of crankpins, trueness of flywheel flange, etc. Dial gages are

Crankshaft Grinding Wheels (S. A. E.)

[1] Nominal Widths, In.	[2] Radii of Edges, In.
1½, 1⅝, 1¾, 1⅞, 2, 2⅛, 2¼, 2⅜, 2½, 2¾, 3, 3¼, 3½, 3¾, 4.	3/32 max.

This specification applies only to unfinished wheels for grinding crankshafts.

[1] Wheels are to be ordered by the normal widths. Wheel manufacturers allow 0.029 in. additional to the nominal widths for truing.

[2] Radii of edges are specified as maximum to permit dressing wheels to desired crankshaft-pin radii.

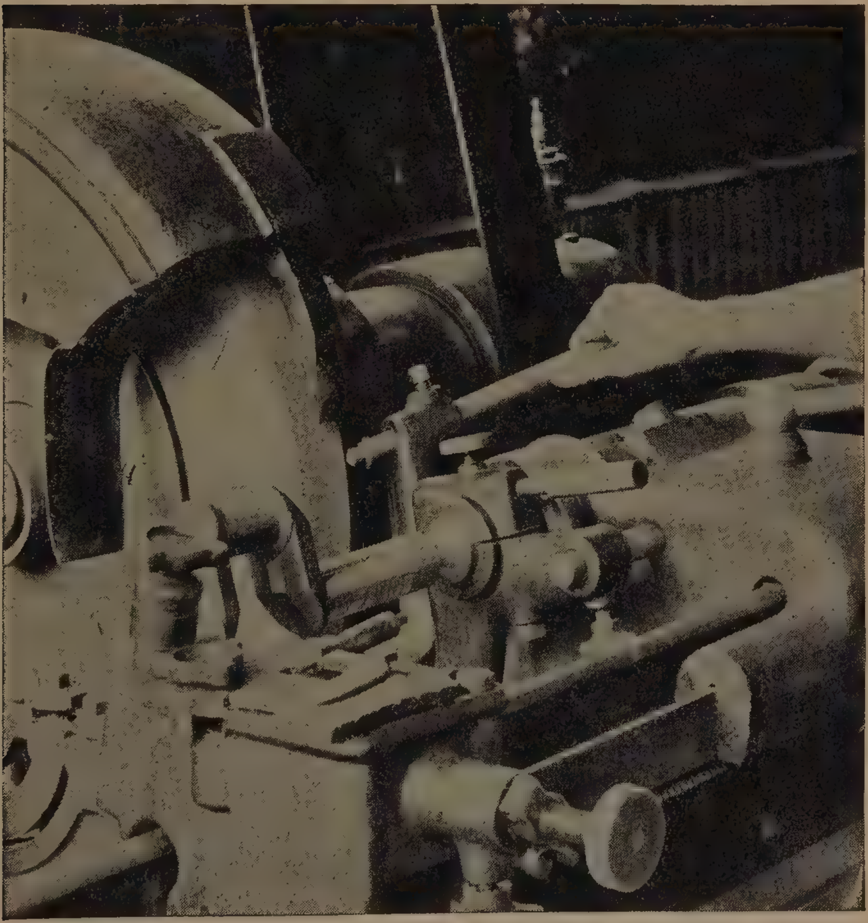


FIG. 144.—SHOWING OPERATION OF TRUING WHEEL BY MEANS OF DIAMOND WHEEL TRUER IN CRANKSHAFT GRINDING WORK.

largely used for these inspections, but use is made also of micrometers.

Balancing Crankshafts—Crankshafts, after they are completely machined, must be balanced both statically and dynamically. A crankshaft is in static balance if, when placed on horizontal steel balance ways or their equivalent, it will remain in any position to which it is turned. A simple static balancing machine consists of two pairs of discs freely supported either on hardened centers or on ball bearings (Fig. 145), the two pairs being sufficiently far apart sc the crankshaft can be placed upon them with its end main bearings. If the center of gravity of the crankshaft does not lie in the mechanical axis, then the crankshaft will turn until the center of gravity is directly underneath the mechanical axis. By removing material from the heavy side, with a drill or emery wheel, until the crankshaft will remain in any angular position in which it is placed on the discs, static balance may be attained.

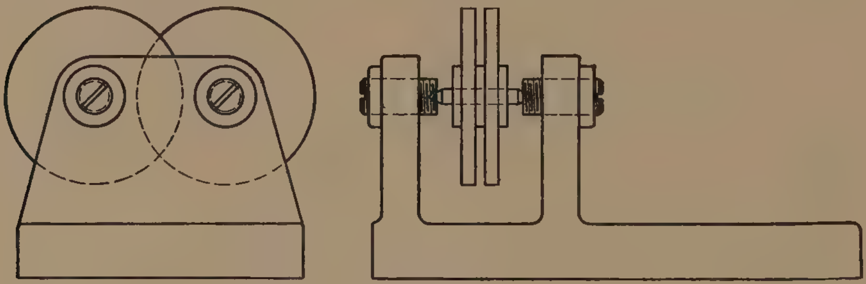


Fig. 145.—Static Balancing Machine.

Dynamically the shaft may still be unbalanced. For instance, there may be excess weight on one side of the shaft at one end, which is balanced statically by an equivalent weight on the other side at the opposite end. In that case, when the engine is running, there is what is known as a centrifugal couple, and this must be eliminated before the crank can be expected to run without vibration at high speeds. Unbalance of this kind can be determined only in a dynamic balancing machine.

Pendock Balancing Machine—A very simple crankshaft balancing device has been developed by C. W. Pendock and is illustrated herewith (Fig. 146). From the line shaft a short vertical shaft is driven by means of a quarter-twisted belt. From this vertical shaft the crankshaft to be balanced is hung through a cotton rope about a foot long, a loose-fitting ball and socket joint and a screw cap screwing over one end of the crankshaft, which happens to be threaded. The whole is driven by a variable-speed electric motor. The crankshaft is brought up to speed gradually, by starting the motor gradually, on account of the ball and socket joint, which will transmit only a slight driving torque.

The crankshaft to which this device has been particularly adapted is a four-cylinder, two-bearing type, and after it has been brought up to speed, a scriber is approached first to one and then to the other main bearing, until a mark is made on same. Before putting the crankshaft into this device it has been carefully balanced on a static balancing machine, so that if it is heavy on one side at one end, it must be also heavy on the other side at the other end. Therefore the scribers will mark the two main bearings on opposite sides.

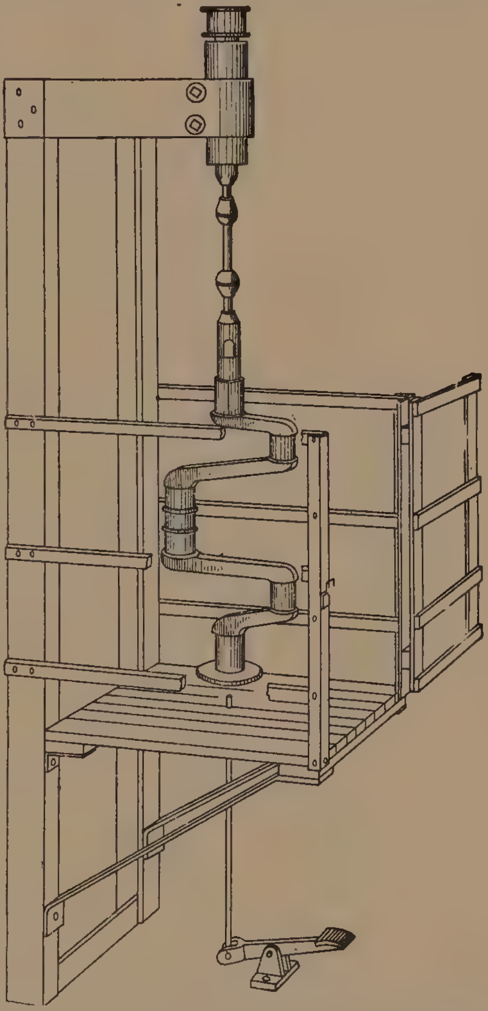
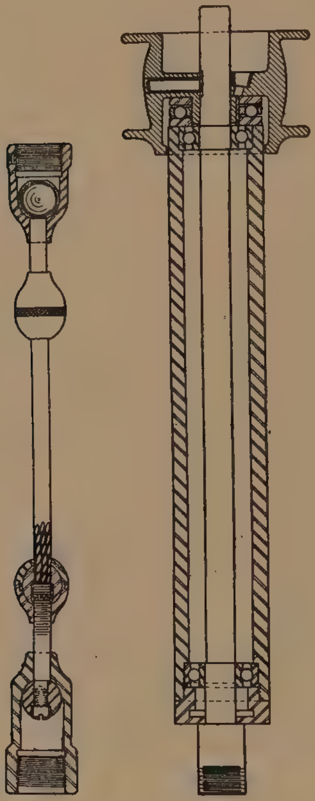


Fig. 146.—Pendock Crankshaft-Balancing Machine.

The sides marked are the light sides. This is due to the fact that when a shaft is thus rotated while freely suspended it will turn around its axis of gyration. Assume a crankshaft that has been statically balanced to be heavy at W and W (Fig. 148). Then its center of gyration will be located as indicated by the line *c-c*, and the points on the two main bearings furthest from the axis of gyration, and which will be marked by the scriber, are *a* and *b*.

This machine does not measure the amount of unbalance and therefore is not so well suited to quantity production as the two machines of which descriptions follow. But it is useful in cases where only a limited number of crankshafts are to be balanced and none of the other machines (which are somewhat expensive) are available.



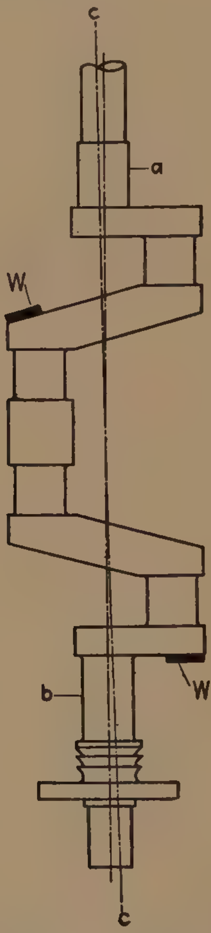


Fig. 147 (left).—Parts of Pendock Balancing Machine.
Fig. 148 (right).—Diagram Showing Axis of Gyration of Unbalanced Shaft.

Precision Balancing Machine—This machine, illustrated in Fig. 149, was invented by Dr. B. L. Newkirk and is being manufactured by the Gisholt Machine Co., Madison, Wis. It consists of a bed on which is supported, by means of two thin vertical spring steel blades, a tubular framework on which the crankshaft can be supported and revolved. The framework is steadied by a horizontal cantilever spring with a scroll at its end.

The stiffness of this spring can be changed by varying its effective length, and there is also a micrometer adjustment between the scroll spring and the outer end of the tubular frame. It will be noticed that one end of the crankshaft is located directly over the "knife-edge" bearing support of the frame, so any vibration induced in the frame is due to the unbalance at the opposite end.

A special type of headstock is carried by the frame, as are adjustable rollers for supporting the crankshafts, etc. If an unbalanced crankshaft is placed on the machine and then rotated, it will set the freely mounted frame in vibration. The frame with any particular crankshaft on it has a natural period of vibration, and if the crankshaft is rotated at such a speed that the speed of rotation corresponds to the natural period of the frame and crankshaft, even a slight degree of unbalance will cause a comparatively heavy vibration. This speed is referred to as the critical speed. The method of operating the machine is as follows:

The crankshaft is placed on the supporting rollers and lightly clamped in a chuck on the face plate. Coupled to the crankshaft are the revolving parts of the headstock, which are perfectly balanced. To take a reading, the system is brought up to a speed slightly above the critical speed of the frame, and allowed to slow down by the slight friction of the ball bearings. As the critical speed of the frame is passed through, there occurs a maximum oscillation, which is directly proportional to the unbalanced moment of the overhanging end of the crankshaft. The maximum amplitude of the vibration is read off on a dial mounted on a standard or instrument column, to which connection is made from the vibrating frame by a steel rod or wire. Except when making tests, the frame is held steady by means of a taper plug on this standard, which enters a corresponding hole in the frame. Best results are obtained with a critical speed of 100-110 r.p.m., and the cantilever spring is adjusted to give this critical speed. The headstock can be spun either by hand or by a small electric motor.

From the maximum amplitude of vibration as read off on the indicator, and a calibration factor separately determined, the unbalanced moment, in ounce-inches, can be directly unascertained. A corrective moment is then applied to the rotating system in the form of a sliding weight on the headstock, which is moved out radially from the axis and clamped in position at such a distance from the axis as to give the necessary moment.

To find the angular position at which the correction is

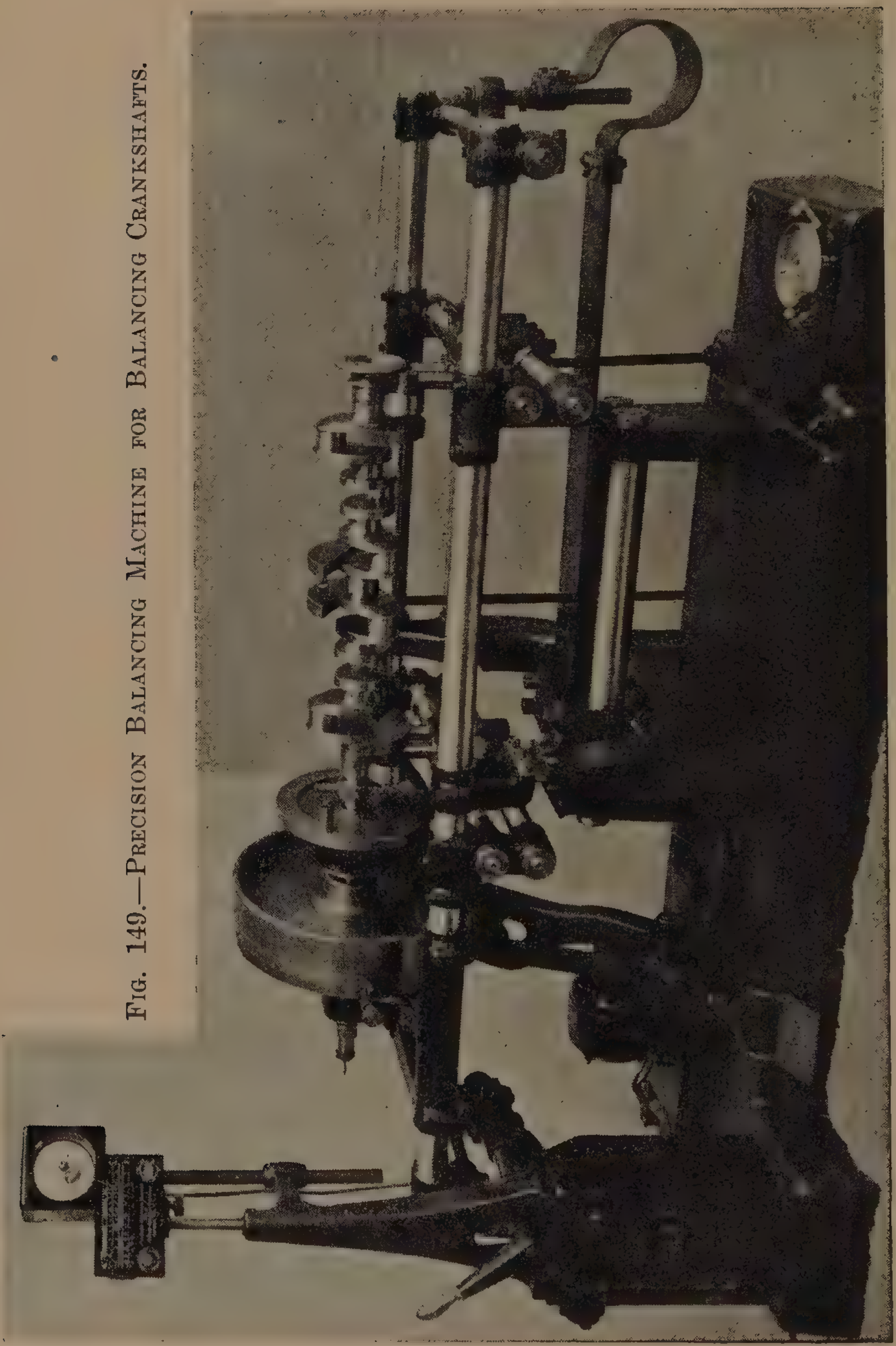


FIG. 149.—PRECISION BALANCING MACHINE FOR BALANCING CRANKSHAFTS.

required, the correcting moment is arbitrarily applied to the headstock at an assumed angle and the system is again speeded up and allowed to slow down through the critical speed. The maximum amplitude this time will be different, the ratio between the two readings being dependent upon the angle betwen the point of application of the corrective moment and the correct point. The trigonometric value of this ratio is

$$R = 2 \cos \tfrac{1}{2} (180^\circ - \alpha),$$

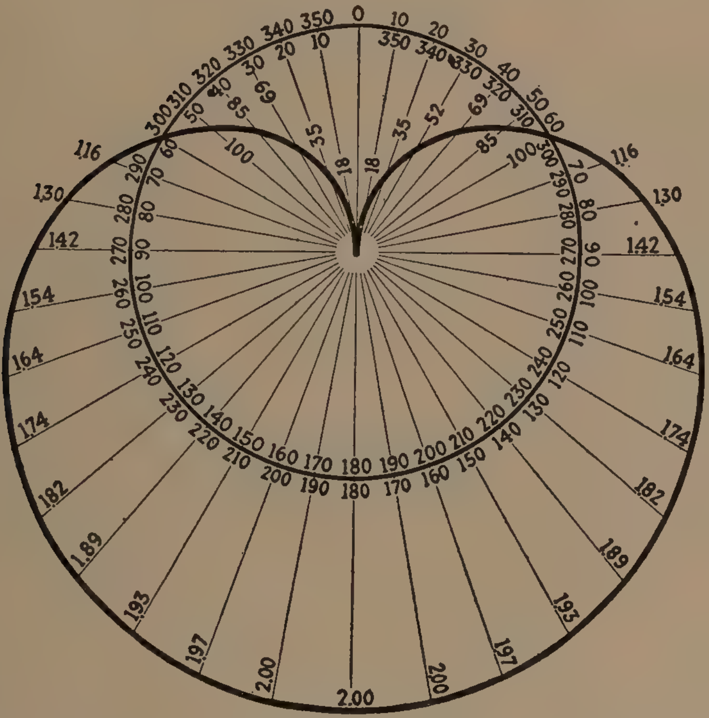


FIG. 150.—DIAGRAM USED WITH PRECISION BALANCING MACHINE FOR FINDING ANGLE OF UNBALANCED MOMENT.

where α is the required angle. This function, plotted on a polar diagram, is shown in Fig. 150, which may be used in determining the angle after first having found the ratio of the two amplitudes. A scale corresponding to Fig. 150, is built into a special slide rule mounted on the instrument column. Scales are also provided on this column for employing the calibration factor and taking ratios, so the operator is relieved of the necessity of making computations. The angle ascertained is set off on the headstock dial and another run is made to check the result.

After the magnitude of the unbalanced moment and its angular position have been determined for one end of the crankshaft, the latter is turned end for end on the roller supports and another determination is made. This machine serves to test a rotating part for both static and dynamic balance.

Carwen Static and Dynamic Balancing Machine—Diagrammatic drawings of the Carwen static and dynamic balancing machine, manufactured by the Tinius Olsen Testing Machine Co., of Philadelphia, are shown in Fig. 151. The diagram shows a rotating body of the type of an armature mounted in the machine, but the same principles apply in balancing a crankshaft. The underlying principle is that of creating an artificial unbalanced moment in a part which rotates at the speed of the part to be tested, to counteract the unbalanced moment of the latter.

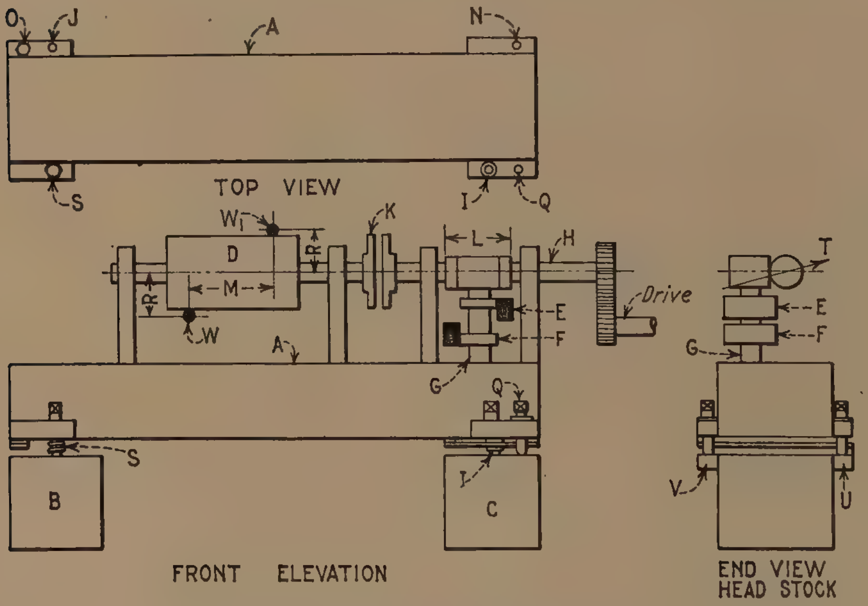


FIG. 151.—DIAGRAMMATIC VIEWS OF OLSEN-CARWEN STATIC AND DYNAMIC BALANCING MACHINE.

The part D to be balanced is mounted in bearings and connected by a flexible coupling K to the driving spindle H. A vibrating bed A is supported from pedestals B and C through the helical springs S, I, and O, and through pivot pins J, N, Q. E and F are balancing blocks mounted on a vertical shaft G, and L is a sliding sleeve by means of which the angular relation of shaft G to shaft H can be changed.

First a test for static unbalance is made. For this purpose the vibrating bed is supported on pivots J and N in such a way that it is free to vibrate in the direction T around a line through the contact points of J and N. Any static unbalance in the body being tested will cause the bed to vibrate around this line, whereas if the body is statically balanced but dynamically unbalanced there will be no vibration. The moment of static unbalance can be measured and its direction found by shifting block F to the proper distance from E on shaft G, and at the same time changing the angular relation between the body D and the vertical shaft G by shifting sleeve L along shaft H in one direction or the other. When vibration ceases the artificially created moment is exactly equal and opposite to the static unbalance of body D. Block F and sleeve L are shifted while the machine is running.

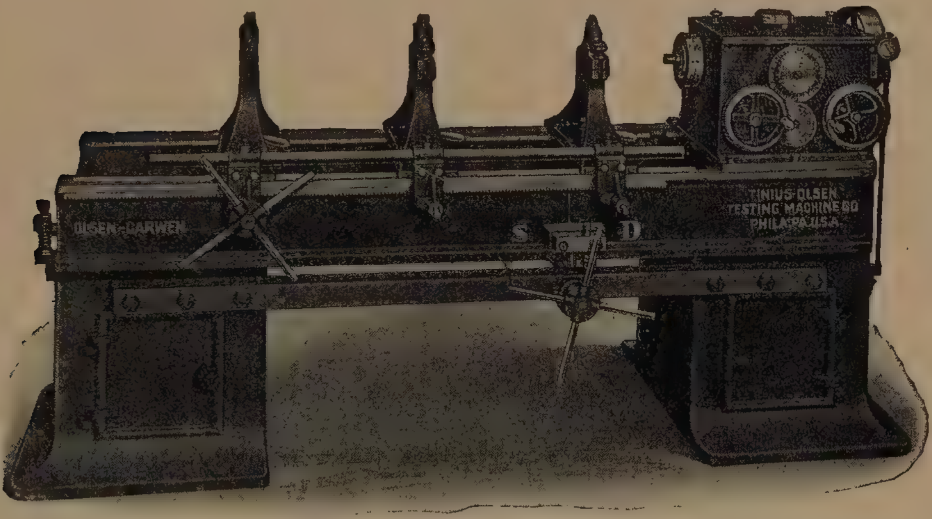


Fig. 152.—Olsen-Carwen Static and Dynamic Balancing Machine.

After the body (crankshaft) has been placed in static balance, a test for dynamic unbalance is made. For this purpose bed A is supported on pivots N and Q and is made free at the opposite end, pivot J being raised so as to be out of contact with pedestal B. If there is any dynamic unbalance—that is, if the rotating body is too heavy on one side at one end and on the other side at the opposite end—there wil be vibration around the line through the points of contact of pivots N and Q. This vibration is stopped by shifting block F and sleeve L, the same as in the test for static unbalance.

The change-over from a static to a dynamic balancing machine is made by unscrewing the spring screw I until pivot Q comes in contact with column C and tightening spring screw O until pivot J is out of contact with column B. To change back to a static balancing machine the reverse operations are performed.

One size of this machine is claimed to have such a degree of sensitiveness that a static unbalance of ½ oz.-in. is registered, and a dynamic unbalance of 1 oz.-in., while the machine will locate the angle of unbalance to within 2°.

On top of the beadstock there is a drum dial which indicates the plane of unbalance for both static and dynamic balance tests. The upper dial on the front of the machine indicates the unbalance in ounce-inches in both tests. As a convenience in rechecking parts that have been corrected by means of this machine, an angle indicator dial is provided at the bottom of the front plate. The hand wheel at the left serves to change the angular position of the counterbalance blocks E and F, while that on the right serves to change the amount of unbalance.

Method of Removing Metal for Balancing—With crankshafts in which the crank arms are left unfinished, the usual practice is to remove metal for balancing purposes by drilling into the crank arms from the end, and some manufacturers of balancing machines have issued tables to show the depth of holes of different sizes required to correct a certain unbalance, taking account of the distance of the holes from the axis of rotation.

With crankshafts that are finished all over this is impractical, as one of the objects of thus finishing them is to obtain an improved appearance, which would be marred if the arms were drilled at points where no holes were intended in the design. The Packard Motor Car Co., who finish their crankshafts all over, remove uniform layers of metal from the outer flat side of the crank arm by grinding.

CHAPTER X

Determination of Bearing Loads

After an engine has been designed, the loads on the bearings under different working conditions can be calculated with a fair degree of accuracy, and it is always well to make these calculations, in order to check the dimensions adopted. For this purpose, the gas pressures in the cylinder during the compression and expansion strokes must be determined from an indicator diagram, and, since no actual diagram can be available until after the engine has been built, one must be drawn from data of the performance of similar engines, taking account of the compression ratio provided for in the design and the volumetric efficiency which may reasonably be expected at the speed for which it is desired to determine the bearing loads. The weights of the reciprocating and unbalanced rotating parts can be calculated from the design.

The forces due to the gas pressure and to the inertia of the reciprocating parts are always in the direction of the cylinder axis, but, owing to the angularity of the connecting rod, and especially on account of the centrifugal forces due to the connecting rod head and of the unbalanced portion of the crankshaft, the loads on the crankshaft bearings, instead of being uni-directional, travel around in a circle, changing in magnitude at the same time. It has become customary to determine the magnitude and direction of these loads for twenty-four points of the cycle, separated by equal angular distances of 30°. The values thus found are used in plotting a polar diagram of bearing loads. Each instantaneous load is laid off on a radial line in the direction in which it acts, the points thus obtained are numbered consecutively from 1 to 24, and a curve is drawn through them (or the points may be connected by straight lines).

Theoretical Indicator Diagram—We will illustrate the method of calculating the bearing loads by an example. For the sake of simplicity we will assume that the compression space consists of an extension of the cylinder beyond the top end of the stroke, the diameter of the extension being the same as the cylinder bore. With the compression space a direct extension of the bore, the volume of the combustion chamber varies as the distance from the top end of the piston to the top of the compression space. We will assume a four-cylinder engine of 3½-in. bore and 5-in. stroke, with a compression ratio of 4½. The engine has a three-bearing crankshaft and is to operate at a maximum speed of 2,500 r.p.m. Since the compression ratio is 4½, the height of the equivalent cylindrical compression space is

$$\frac{5}{4.5 - 1} = 1.428 \text{ in.}$$

We next find the distance of the piston head from the top end of the stroke for crank angles of 30°, 60°, 90°, 120° and 150° by means of the equation

$$x = \frac{l}{2}(1 - \cos\theta) + nl\left(1 - \frac{\sin^2\theta}{4n^2}\right).$$

Assuming the connecting rod length l to be equal to 2.25 times the length of the stroke, and substituting this value of n and the value of l (5 in.) in the above equation we get

$$x = 2.5(1 - \cos\theta) + 11.25\left(1 - \frac{\sin^2\theta}{20.25}\right).$$

By means of this equation we find the distances the piston has moved down the cylinder for the different crank positions. We next add to these distances the height of the compression volume, and then, by dividing the sums thus obtained by the height of the compression volume, we find the volume ratios in which the gases have expanded during the various crank motions. By raising these ratios to the 1.3 power we find the ratios in which the pressure has decreased during the various crank motions, and dividing the initial pressure (the theoretical pressure in the cylinder at the beginning of the expansion stroke) by the latter ratios, we obtain the pressures in the cylinder after the crank has turned through the various angles.

Compression pressures, or, rather, pressures in the cylinder at various points of the compression stroke, are calculated in the same way. We first calculate the maximum compression pressure, which is reached at the end of the compression stroke, and from this we find the pressures at other points of the stroke by dividing by the same ratios as in the case of the expansion stroke.

At the beginning of the compression stroke the pressure in the cylinder at an engine speed of 2,500 r.p.m. is in the neighborhood of 12 lbs. per sq. in. abs. Since the compression ratio is 4.5 and the exponent of the compression curve is 1.3, the maximum compression in the cylinder at the end of the compression stroke is

$$12 \times 4.5^{1.3} = 85 \text{ lbs. per sq. in. abs.}$$

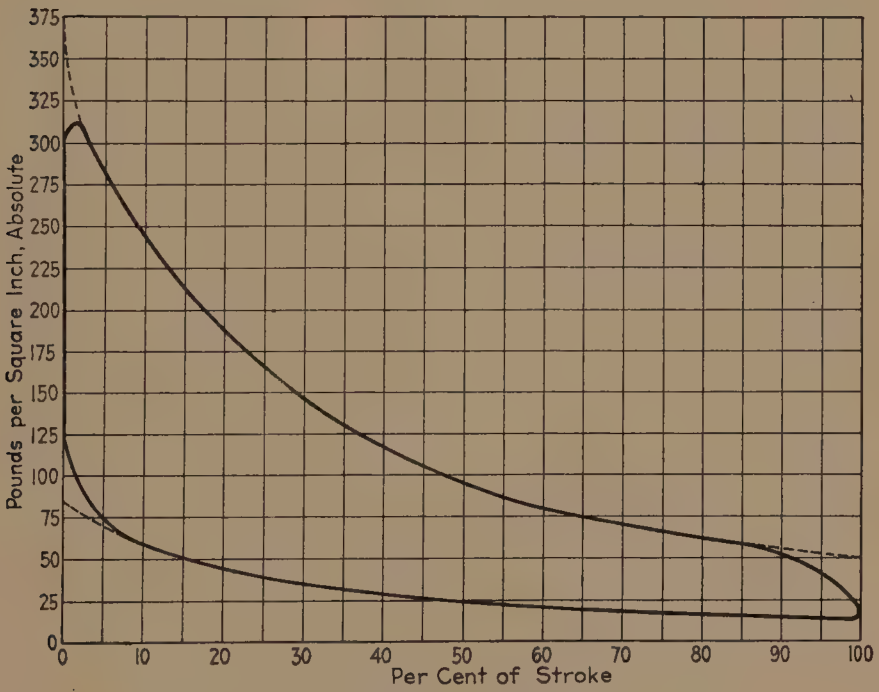


FIG. 153.—THEORETICAL INDICATOR DIAGRAM.

The pressure multiplication on explosion for a 4.5 compression ratio is about 4.25, and the theoretical maximum pressure in the cylinder therefore is

$$4.25 \times 85 = 361 \text{ lbs. per sq. in. abs.}$$

The various calculations outlined in the foregoing have been carried through for the case in hand and the results are given in Table VII:

Next the pressures are reduced to gage pressures by subtracting 14.7 from them, and then the total pressures on the piston are found for any crank position by multiplying

Table VII—Variations of Gas Volume and Pressure

Crank Position	Piston Displacement	Volume Ratio	$v^{1.3}$	Explosion Pressure Abs.	Compression Pressure Abs.
0°	.....	.....	....	361	85
30°	0.406	1.286	1.39	259	61
60°	1.460	2.023	2.50	144	34
90°	2.782	2.950	4.08	88.5	20.8
120°	3.960	3.775	5.64	64.0	15
150°	4.736	4.320	6.71	53.8	12.6
180°	5.000	4.500	7.08	51.0	12

by the piston head area, 9.62 sq. in. From the figures thus obtained the theoretical pressure diagram Fig. 153 can be drawn, and after this has been done the sharp corners are taken off to reduce the area of the diagram by from 5 to 6 per cent, the rest of the loss allowed for in the diagram factor (0.9) being due to back pressure during the suction and exhaust strokes.

Inertia Forces—Since the engine is to operate at high speed, the reciprocating parts must be made very light, and we will assume that their weight per cylinder is 3 lbs. The total inertia force for any crank position can then be found from the equation

$$F = \frac{WlN^2}{70480}\left(\cos\theta + \frac{1}{2n}\cos 2\theta\right).$$

Inserting the known values we get,

$$F = (3 \times 5 \times 2500 \times 2500/70480)\left(\cos\theta + \frac{1}{2n}\cos 2\theta\right)$$

$$= 1331\left(\cos\theta + \frac{1}{2n}\cos 2\theta\right).$$

The values of the term in parentheses can be obtained from the table of crank angle factors for inertia force.

In determining the total forces on the piston, as well as the loads on the bearings, it is important that each force be preceded by its proper sign. In accordance with the usual practice, we call all upward forces and forces toward the right, positive forces, and those contrarily directed, negative. In the following table are given the unit pressures and also the total pressures in the cylinder during the expansion and compression strokes. Allowance is made in this table for the changes in the cylinder pressures at the ends of the strokes due to the rounding off of the diagram at the corners. The table also contains the total inertia pressures.

The next step consists in tabulating the pressures in the direction of the cylinder axis, that is the algebraic sums of the gas pressure and the inertia force, for the twenty-four points of the cycle. In doing this it is convenient to start with the beginning of the inlet stroke. This tabulation for the case under consideration is given below.

Table VIII—Gaseous and Inertia Pressures during Expansion and Compression Strokes

Crank Position	Expansion Stroke Pressure Gage	Compression Stroke Pressure Gage	Total Expl. Press. Lbs.	Total Comp. Press. Lbs.	Total Inertia Force Lbs.
0°	346	70.3	—2,692	—676	1,624
30°	244	46.3	—2,350	—445	1,298
60°	130	19.3	—1,250	—186	518
90°	74	6.1	—712	—59	—295
120°	49	0.3	—473	—3	—813
150°	39	—2.1	—375	+20	—1,005
180°	36	—2.7	—346	+26	—1,035

Table VIII-A—Resultant of Total Gaseous and Inertia Pressures

Inlet Stroke	Lbs.	Power Stroke	Lbs.
0°	1,624	360°	—1,068
30°	1,298	390°	—1,052
60°	518	420°	—732
90°	—295	450°	—1,007
120°	—813	480°	—1,286
150°	—1,005	510°	—1,380
180°	—1,035	540°	—1,381

Compression Stroke	Lbs.	Exhaust Stroke	Lbs.
180°	—1,009	540°	—1,035
210°	—985	570°	—1,005
240°	—816	600°	—813
270°	—354	630°	—295
300°	332	660°	518
330°	853	690°	1,298
360°	948	720°	1,624

Forces Acting Along Connecting Rod—From this point on the work is preferably done graphically, except for the fact that the value of the centrifugal force, which is constant throughout the cycle, is determined by means of a formula. A diagram is drawn of the piston, connecting rod and crank circle, indicating by radial lines the crank in positions 30° apart, and the connecting rod, represented by a straight line, in corresponding positions. From the top end of the connecting rod, which, of course, is on the cylinder axis, is dropped a vertical line, representing to a certain scale the resultant force in the direction of the cylinder axis for that particular crank position, and from the lower end of this line a horizontal line is drawn to the connecting rod center line. The distance along the connecting rod line to the point of intersection with the horizontal line then represents to the same scale the force acting along the connecting rod, the force on the rod being upward if the effective force in the cylinder axis is positive and downward if it is negative. The direction of the forces along the connecting rod is then indicated by arrow heads. It is best to draw two diagrams of this kind (Figs. 154 and 155) one for each revolution of the cycle. The forces along the connecting rod are identical with those on the piston pin bearing, except for the small additional loads due to the whipping effect on the rod, which may be neglected. The load on the piston pin bearing therefore is always in the direction of the connecting rod.

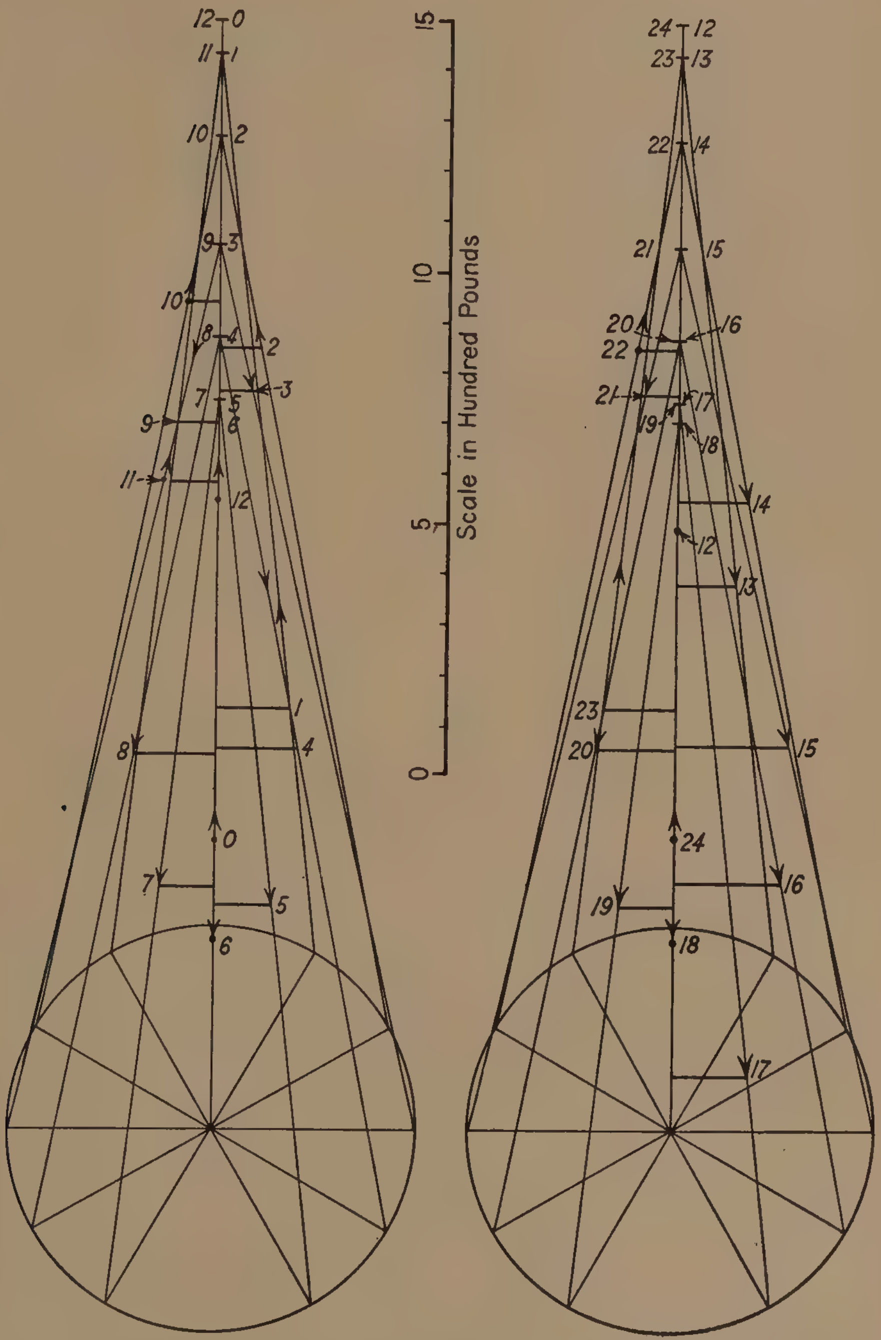
In determining the loads on the crankpin bearing we must first calculate the centrifugal force on the lower or rotating portion of the connecting rod. We will assume this portion of the rod to weigh 1.5 lbs. The centrifugal force is calculated by the equation

$$F_c = \frac{Wn^2r}{35,240} \text{ lbs.},$$

where W is the weight of the rotating mass in pounds; r, its average radius in inches, and n the number of revolutions per minute. We have in this case,

$$F_c = \frac{1.5 \times 2500 \times 2500 \times 2.5}{35,240} = 665 \text{ lbs.}$$

This force always acts in the direction of the crank arms. The next step consists in combining this force along the crank arm with the force along the connecting rod for any crank position. This is done by means of the parallelogram of forces, and the result is the load on the crankpin bearing corresponding to the crank position. In Fig. 156 this operation is carried out for crank position 1, that is 30° after the top dead center at the beginning of the inlet stroke.



FIGS. 154 AND 155.—FORCES ACTING ALONG CONNECTING ROD AT TWENTY-FOUR EQUIDISTANT POINTS OF CYCLE.

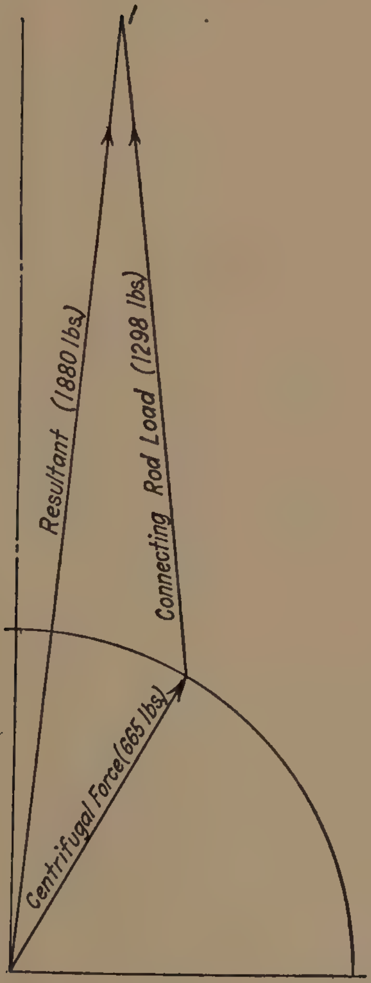
Crankpin Bearing Loads—Similar operations have been carried through for all of the twenty-four crank positions, and from the results obtained the diagram Fig. 157 has been drawn, which represents the load on the crankpin bearing throughout a cycle or two crankshaft revolutions. It will be noticed that the crankpin bearing load revolves at almost, but not exactly, the same rate as the crank itself. For instance, at the beginning of the cycle both the crank and the crankpin bearing load are directed vertically upward, while in position 3, after the crank has turned through an angle of 90° and therefore extends horizontally to the right, the crankpin bearing load already has passed the 90° position and extends downwardly.

Since both the crankpin and the load on it make a half revolution in the same time and never vary appreciably in rotative speed, the load on the crankpin is always on the same side—the inner side—deviating only slightly from a line parallel to the crank arms. It will be noticed that the load on the crankpin bearing attains a maximum value at the end of the exhaust and the beginning of the inlet stroke. The effect of the explosion in the cylinder is to change the direction of the load on the crankpin from upward to downward, and the load is then on the outside of the crankpin for an instant.

Main Bearing Loads—Next the load on the main bearings of the crankshaft is determined. Each of the end bearings takes one half of the load on the adjacent crankpin bearing, and in addition the load due to the centrifugal force on one half of the adjacent crankpin and on the unbalanced portion of the adjacent crank arm. The average radius of the unbalanced portion of the crank arm is practically the same as the crank radius (2½ in.) and the total weight of this unbalanced rotating mass (short crank arm and one half of crankpin) may be assumed to be 2.25 lbs. This, by the use of the same formulae as above, gives a centrifugal force of 1000 lbs.

For each crank position this centrifugal force in the direction of the crank arms is now combined, by the parallelogram of forces, with one half of the corresponding crankpin bearing load, and from the twenty-four values thus obtained the bearing load curve for the front and rear main bearings (the inner curve in Fig. 158) is drawn. The diagram is somewhat ‘‘fuller’’ or more nearly circular than the diagram of crankpin bearing load, which is due to the fact that in this case centrifugal force plays a more important part.

The center main bearing carries one half of the load on each of the two adjacent crankpins, as well as the load due centrifugal forces on the unbalanced portions of the two adjacent crank arms and on one half of each of the two adjacent crankpins. In constructing the bearing load diagram for the center bearing it must be borne in mind that while the cylinder to one side of the center bearing is going through the first half of the cycle, that on the other side is going through the last half. Thus, while the crank



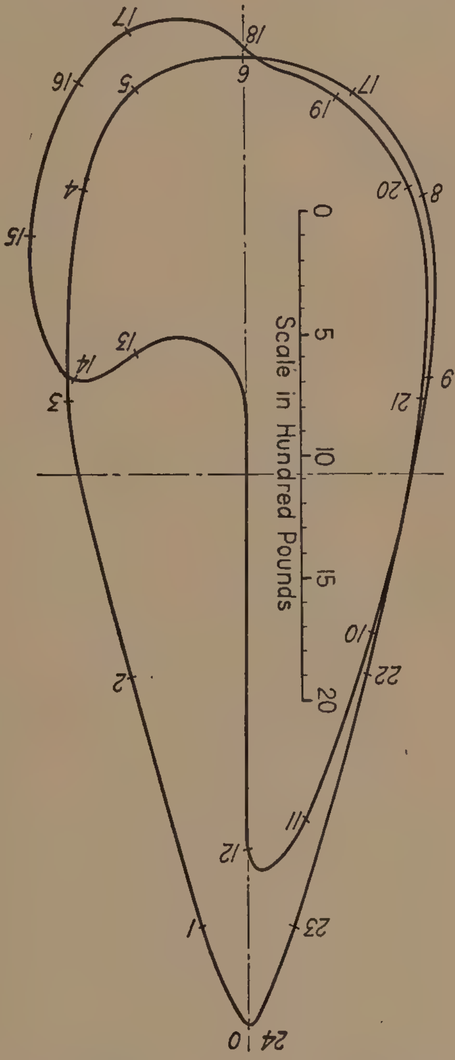


FIG. 156 (LEFT).—COMPOSITION OF CRANKPIN LOAD 30° AFTER BEGINNING OF INLET STROKE.
FIG. 157 (RIGHT).—POLAR DIAGRAM OF CRANKPIN LOAD.

on one side of the bearing is in position 1 that on the other side is in position 13; while the first is in position 2, the other is in position 14, and so on. It is then only necessary to combine the bearing loads found for crank positions 1 and 13, 2 and 14, etc., respectively, by means of the parallelogram of forces, and from the twelve values thus obtained construct the bearing load diagram for the center bearing (the outer curve in Fig. 158).

It will be noticed in the first place that this load passes through a complete cycle during every revolution of the crankshaft. Also, that its instantaneous values are about twice as great as those of the loads on the front and rear main bearings. All of the main bearing loads also make one half revolution during the time of one half crankshaft revolution, and the load therefore always remains on the same side of the crank journal, deviating only slightly to both sides of a central plane.

That portion of the main bearing loads which is due to centrifugal forces can be eliminated by providing the crank arms with suitable balance weights. In the above case we assumed that the rotating portion of the connecting rod weighed 1½ lbs. and that the unbalanced portion of a short crank arm and one half the adjacent crankpin combined weighed 2¼ lbs. Only one half of the centrifugal force on the rotating part of the connecting rod is transferred to one crank arm, hence the total rotating weight for which counterbalance must be provided on one crank arm is 3 lbs., and if the average distance of the balancing mass from the crankshaft axis is equal to the crank radius, the balancing mass on each arm must weigh three pounds.

If the crankshaft is thus balanced, the load on the front or rear main bearing for any crank position is equal to one half the force along the connecting rod for that position and may be found from Figs. 154 and 155. For the center bearing the load for crank position 1 is equal to the mean between the forces along the connecting rod for crank positions 1 and 13; for crank position 2 it is equal to the mean between the forces along the connecting rod for positions 2 and 14, and so on. If the reciprocating and unbalanced rotating weights are related to each other as in this case (which is normal) the maximum instantaneous bearing loads can be reduced by about one half.

This method of analysis may be applied to the determination of bearing loads in engines with any number of cylinders. If there are two or more throws between supporting bearings, the load on any crankpin is divided between the adjacent main bearings in the inverse proportion of their distances from the crankpin (center-to-center distances). It may also be used for determining bearing loads under conditions of partial engine load.

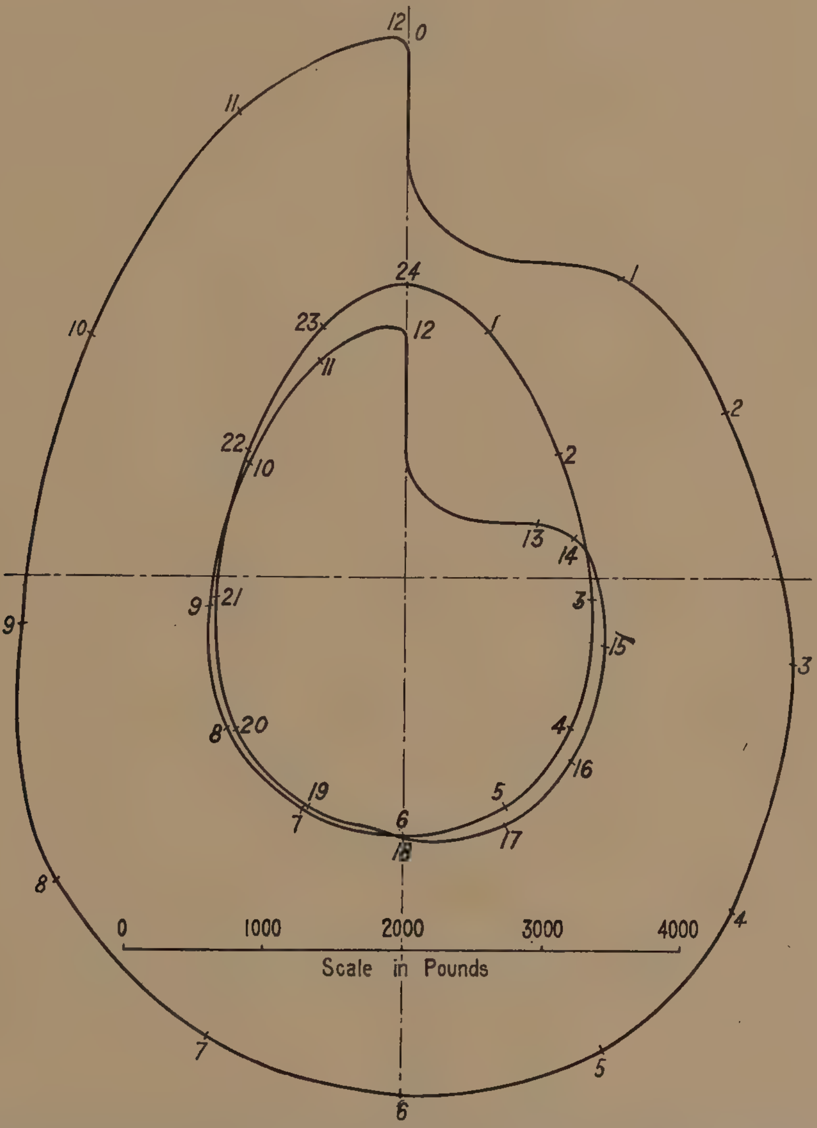


Fig. 158.—Polar Diagrams of Load on Front and Rear Bearings (Inner Double Loop) and on Center Bearing (Outer Loop).

The load on the rear bearing is somewhat affected by the weight of the flywheel and clutch. The load due to this cause is substantially equal to the weight of these parts and is directed downwardly, hence can be readily taken account of in the diagram. In the example above worked out the maximum load on the rear bearing is about 2250 lbs. and as the flywheel and clutch of such an engine would certainly not weigh over 100 lbs., they do not change the bearing load much. This weight decreases the maximum upward and increases the maximum downward load, and as the former is the larger of the two, at high engine speed the effect of the flywheel weight is really to decrease the maximum bearing load.

CHAPTER XI

The Connecting Rod

The connecting rod is the intermediate member between the crankshaft and the piston. It consists of the shank, the head or big end, which forms a bearing for the crankpin; and the small end, which forms a bearing for the piston pin or has the latter clamped in it.

Connecting rods are almost invariably drop-forged from either medium carbon (about 0.35 per cent) or chrome nickel steel. The former may be assumed to have a tensile strength of 95,000 lbs. per sq in.; the latter of 125,000. In recent years duralumin, a forgeable aluminum alloy, has come into use for connecting rods to a certain extent. The properties of this alloy will be discussed further along in this chapter, together with points in the design of rods to be made of it.

Length of Rod—In determining the dimensions of the rod we have to consider, first, the necessary length, and, second, the necessary cross-section. In speaking of the length of the connecting rod, the distance between the axis of the piston pin and the axis of the crankpin bearing is referred to, which is known as the center-to-center length. This length is generally expressed in terms of the stroke. In our discussion of the crank moment and of side thrust on the cylinder wall we assumed different values for the ratio between the length of the connecting rod and the length of stroke, ranging between 1.75 and 2.5.

For an engine with offset cylinders and intended for a purpose where lightness is paramount, the smaller ratio can be chosen. In American passenger-car engines of the symmetrical type the ratio generally lies between 2 and 2.25. European designers in the past sometimes have provided a connecting rod equal to 2.5 times the length of the stroke, and this ratio can be recommended for commercial vehicle engines, in which long life is more important than low weight.

A consideration that influences the length of the rod is that it must clear the bottom end of the cylinder bore when in its position of greatest angularity. The cylinder, crank and connecting rod are laid off on the drawing board, as shown in Fig. 159, with the connecting rod at right angles to the crank arm. If a certain length of connecting rod does not clear properly, a longer rod may be chosen and the layout worked over again. Sometimes, when the shortest possible connecting rod is wanted, or when the stroke-bore ratio is unusually large, the bottom end of the cylinder is formed with slots into which the connecting rod swings at the moments of its greatest angularity.

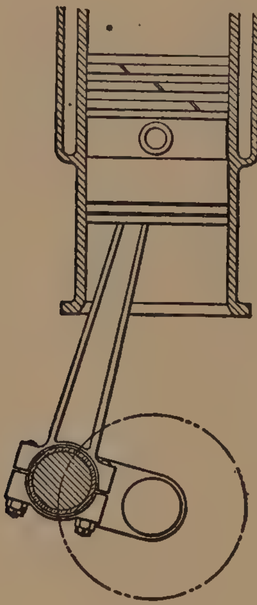


Fig. 159.

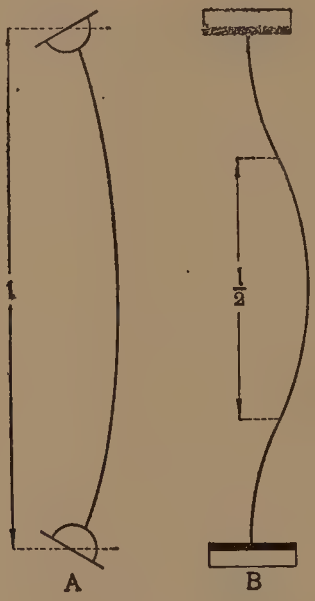


Fig. 160.

Calculation of Cross-section—The connecting rod works chiefly under compression, and must be treated as a column. The most accepted formula for the strength of columns seems to be Rankine's, which is as follows:

$$\frac{P}{A} = \frac{S_c}{1 + q\frac{l^2}{r^2}}, \qquad (61)$$

where P is the total pressure on the column; A, the cross-sectional area at the section of maximum stress; S_c, the working stress; l^2 the square of the length of the column; r^2 the square of the least radius of gyration, and q a coefficient depending upon the condition of the ends of the column (whether fixed or free) and upon the material. The square of the least radius of gyration is found by dividing the moment of inertia of the section by its area. The left hand member of the equation evidently represents the compression per unit of area of the cross-section of the column, and the term $q(l^2/r^2)$ in the denominator of the right-hand member of the equation takes account of the additional stress due to the bending of a long column. When the length of the column is small in comparison with its cross-section this factor is negligible, and the maximum stress is simply P/A.

In the discussion of columns with respect to their strength, a distinction is made between columns fixed at their ends and those free at their ends, the latter being not nearly as strong as the former, if the length is large in comparison with the cross-section. Now, the connecting rod presents a case intermediate between a column with fixed ends and a column with free ends, since it is free at its ends in the plane of motion, and fixed perpendicular to the plane of motion. The result is that if bending occurs in the plane of motion it will be as shown on an exaggerated scale at *A* in Fig. 160, the entire length of the rod tending to form a simple curve concave to the normal center line of the rod. If bending occurs at right angles to the plane of motion, the rod will assume a shape similar to that shown at *B*, Fig. 160. The end portions of the rod will form curves convex to the normal center line, and the center portion a curve concave thereto. At the points where the convex and concave portions join there is a neutral section, at which there is no bending stress. The distance between these two inflection points is equal to one-half the total length of rod, and since the stress necessary to cause a certain bending strain varies as the square of the length of the curve, it follows that the bending moment due a certain pressure on the rod is four times as great in the plane of motion as normal to the plane of motion.

In a connecting rod we have, however, an additional moment tending to cause the rod to bend in the plane of motion which is not present in a stationary column—namely, the moment due to the transverse acceleration of the mass of the rod, or the whipping effect, as it is generally called. But at the instant when the greatest pressure acts on the connecting rod, the lower end of the rod moves transversely at a uniform speed, and the transverse accelerating force is zero. This effect, therefore, may be neglected.

The formula quoted above, viz.:

$$\frac{P}{A} = \frac{S_c}{1 + q\frac{l^2}{r^2}},$$

may be written

$$\frac{P}{A} + q\frac{l^2}{r^2}\frac{P}{A} = S_c, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (62)$$

which shows that the actual maximum stress on the fibers of the rod is made up of two items, the first, P/A, being that due to the compression of the rod and the second, $q\ (l^2/r^2)\ (P/A)$, being that due to the bending moments.

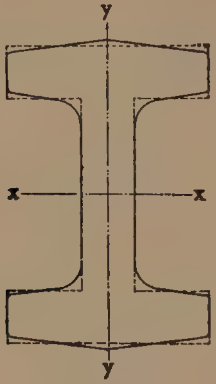


FIG. 161.

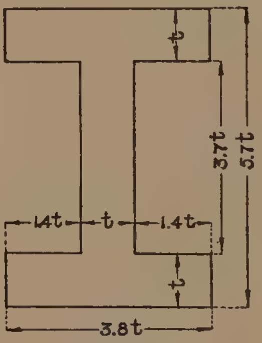


FIG. 162.

From a mass of practical data at hand the writer has calculated the proper values for q and for S_c in equation (62). The value of q is 0.000526, considering the square of the least radius of gyration around the axis x-x, Fig. 161, and that of S_c, 18,000 in the case of carbon drop forge steel and 24,000 in the case of chrome nickel steel, both steels being properly heat-treated.

Hence we may write

$$\frac{P}{A} + 0.000526\frac{P}{A}\frac{l^2}{r^2} = 18{,}000, \quad . \quad . \quad . \quad . \quad (63)$$

for carbon steel and

$$\frac{P}{A} + 0.000526\frac{P}{A}\frac{l^2}{r^2} = 24{,}000, \quad . \quad . \quad . \quad . \quad (64)$$

for chrome nickel steel.

The section of the connecting rod is generally of I shape and the flanges must be given the usual draft. The writer has found it convenient, in calculating the dimensions of the section, to base the calculations on an equivalent section with web and flanges of uniform thickness, that is, without draft. An analysis of sections in actual use showed that the average proportions are about as follows: Calling the thickness of the web and the flange t, the width of the section is $3.8t$ and the height is $5.7t$. The height is therefore, one and one-half times the width. The moment of inertia around the axis x-x (Fig. 161) is

$$I_x = \frac{3.8t(5.7t)^3 - 2.8t(3.7t)^3}{12} = 46.83t^4,$$

and the moment of inertia around the axis is y-y,

$$I_y = \frac{2t(3.8t)^3 + 3.7t(t^3)}{12} = 9.45t^4.$$

The area of the section is

$$(3.8t \times 5.7t) - (3.7t \times 2.8t) = 11.3t^2.$$

Since the square of the least radius of gyration around any axis is equal to the moment of inertia around that axis divided by the area of the section, we have for the square of the least radius of gyration around x-x

$$\frac{46.83t^4}{11.3t^2} = 4.14t^2,$$

and the square of the least radius of gyration around y-y,

$$\frac{9.45t^4}{11.3t^2} = 0.84t^2.$$

Having thus found expressions for A and r_x^2 in terms of t, we may substitute these values in equations (63) and (64) and solve for t. Equation (63) becomes

$$\frac{P}{11.3t^2} + 0.000526 \frac{P}{11.3t^2} \frac{l^2}{4.14t^2} = 18{,}000,$$

$$4.14Pt^2 + 0.000526Pl^2 = 842{,}076t^4,$$

$$t^4 - \frac{Pt^2}{203{,}800} = \frac{Pl^2}{1{,}600{,}000{,}000},$$

$$t^2 - \frac{P}{407{,}600} = \sqrt{\left(\frac{P}{407{,}600}\right)^2 + \frac{Pl^2}{1{,}600{,}000{,}000}},$$

$$t = \sqrt{\sqrt{\frac{P^2}{167{,}000{,}000{,}000} + \frac{Pl^2}{1{,}600{,}000{,}000}} + \frac{P}{407{,}600}} \text{ in.} \quad (65)$$

Having once found the value of t, we can find the other dimensions from the proportions given in Fig. 162. This same method can be applied if the web and flanges are not of the same average thickness. In that case t can be used

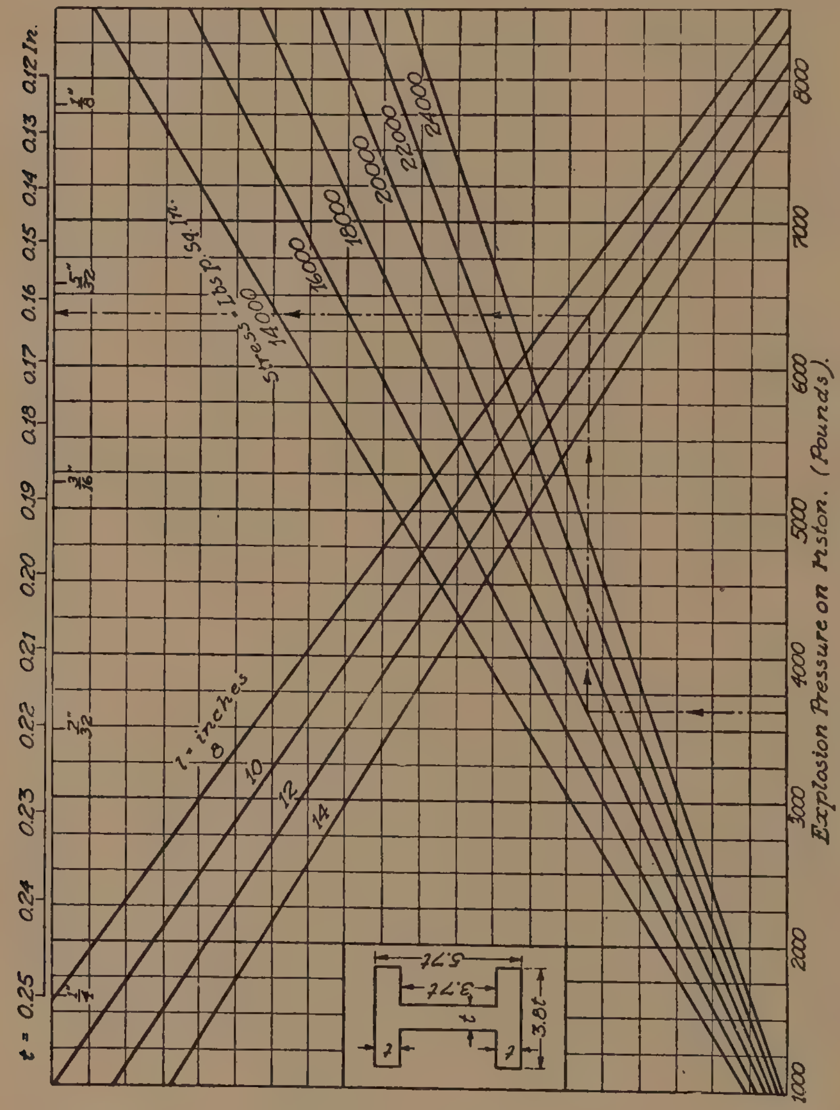


CHART III.—FOR DETERMINING REQUIRED DIMENSIONS OF I-SECTION.

to designate the thickness of the web, and the average thickness t_1 of the flanges can be expressed as a multiple thereof, the same as the width and height of the section. It is fitting to point this out, in view of the fact that there is a tendency at the present time to reduce the thickness of the web, because the metal in the web contributes least to the stiffness of the rod. Equation (65), of course, is applicable only to a section of the proportions shown in Fig. 162.

This equation (65), which gives the necessary thickness of web and flanges for a section of a carbon drop forge steel connecting rod of length l and subjected to an explosion pressure P, is rather inconvenient to use on account of the big fractions involved, and for this reason Chart III has been drawn from which the proper value of t for any values of P and l can readily be determined. The chart permits of finding the proper section thickness for stresses between 14,000 and 24,000 lbs. per sq. in. and even for stresses beyond these limits since stress and load are directly proportional.

Instead of proceeding in the manner outlined, one may choose any section thought to be of desirable form, determine its area and least radius of gyration by the methods explained in textbooks on geometry and theoretical mechanics, and then, by substituting the values thus found in equations (63) or (64), see what load the rod will support. The section can then be increased or reduced proportionately.

Straight vs. Tapering Rods—The section thus calculated is to be used at the middle of the length of the rod, where, at the moment of explosion, the bending stress is the greatest. It is the usual custom to taper the rod, letting the height of the section decrease more or less rapidly from the head down to the small end. There is no apparent reason why this should be done, since the maximum stress occurs midway between the ends. The theoretically correct way would be to make the section largest at the middle and taper it slightly toward the ends, but the gain would be so slight that it would not pay for the more complicated die. The writer considers a straight rod of uniform section from end to end the most practical form. But, as stated, the usual practice is to taper the rod from the head end down to the small end.

Tubular Connecting Rods—Where the reciprocating parts must be of the lowest possible weight, as in racing and other high-speed engines, the connecting rods are sometimes made tubular; that is, the shank is forged with a solid round section and then drilled out from one end and turned off on the outside. Being machined all over, the section will be more nearly uniform than in a rough drop forging, and, therefore, a somewhat higher unit stress can be allowed. In some racing engines with tubular connecting rods of chrome-nickel steel, heat treated, the stress in the connecting rod shank was figured by the author to be in the neighborhood of 36,000 lbs. per sq. in. Most designers however, keep well within this mark, and a stress of 24,000 lbs. per sq. in. would seem to be a good average figure for the shank of a tubular connecting rod of heat treated alloy steel. The necessary diameters may then be calculated as follows: Suppose that the internal diameter is to be three-quarters the external diameter. Then the moment of inertia of the section

$$I = \frac{\pi[D^4 - (\frac{3}{4}D)^4]}{64} = 0.0335D^4,$$

and the area of the section

$$a = \frac{\pi}{4}(D^2 - \tfrac{9}{16}D^2) = 0.344D^2.$$

Hence the square of the least radius of gyration,

$$r^2 = \frac{I}{a} = \frac{0.0335D^4}{0.344D^2} = 0.0974D^2.$$

Inserting in equation (62)

$$\frac{P}{0.344D^2} + 0.000526\frac{P}{0.344D^2} \times \frac{l^2}{0.0974D^2} = 24{,}000$$

$$0.0974PD^2 + 0.000526Pl^2 = 804D^4$$

$$D^4 - 0.000121PD^2 = 0.000000654Pl^2$$

$$D^2 - 0.0000605P = \sqrt{(0.0000605P)^2 + 0.000000654Pl^2}.$$

$$D = \sqrt{\sqrt{(0.0000605P)^2 + 0.000000654Pl^2} + 0.0000605P}$$

$$= \sqrt{\sqrt{\left(\frac{P}{16{,}550}\right)^2 + \frac{Pl^2}{1{,}530{,}000}} + \frac{P}{16{,}550}}.$$

As an example let us take the case of a 3 × 4¾-inch engine. The piston head area is 7.06 sq. in., and if we figure on a maximum explosion pressure of 400 lbs. per sq. in., the pressure on the piston is

$$P = 2{,}824\ lbs.$$

If the connecting rod is 10 inches long between centers we get for the necessary outside diameter of the shank

$$D = \sqrt{\sqrt{\left(\frac{2824}{16{,}550}\right)^2 + \frac{2824 \times 100}{1{,}530{,}000}} + \frac{2824}{16{,}550}} = 0.624 \text{ in.}$$

The inside diameter should be three-fourths of this or

$$\tfrac{3}{4} \times 0.624 = 0.468 \text{ in.}$$

Round figures would be chosen — 5/8 in. for the outside and 15/32 for the inside diameter. A design of a tubular connecting rod is shown in Fig. 169.

Whipping Effect—It is of interest to investigate the "whipping effect" in a connecting rod, which has evidently a considerable influence on the stress in the rod when the the piston is near mid-stroke. In Fig. 163, let A represent the momentary position of the crankpin axis and $B\ C$ the linear speed of the crankpin. If we call the number of revolutions per minute N, then the linear speed is

$$\frac{\pi r N}{30 \times 12}\ \textit{feet per second.}$$

The linear speed can be resolved into two components, the vertical component $O\ B$ and the transverse component $O\ C$. It is the latter component that interests us here. But

$$O\ C = B\ C \cos B\ C\ O = B\ C \cos B\ O\ A,$$

because $O\ A$ is perpendicular to $B\ C$ and $O\ B$ perpendicular to $O\ C$, hence angles $B\ C\ O$ and $B\ O\ A$ are identical. Substituting the value of $O\ C$ and denoting angle $B\ O\ A$ by θ as in former discussions, we have for the transverse component of the crankpin velocity

$$v = \frac{\pi r N}{360} \cos \theta.$$

To find the transverse acceleration we differentiate this expression with respect to time, which gives

$$\frac{dv}{dt} = - \frac{\pi r N}{360} \sin \theta \frac{d\theta}{dt}.$$

But

$$\frac{d\theta}{dt} = \frac{\pi N}{30},$$

hence

$$\text{acceleration} = \frac{\pi^2 N^2 r}{10{,}800} \sin \theta \text{ feet per second per second.}$$

Now, it is only the shank portion of the rod that causes a whipping effect, since the two hubs are symmetrical around their journals and therefore balance around the supports, as it were. But we will not be far wrong in assuming that the rod extends from center to center of bearings. This will give a result slightly too large, so any calculation of dimensions based upon it will be on the safe side.

The above expression for the acceleration attains its maximum values when θ is 90° and 270°, when $\sin\theta$ is equal to 1 and -1, respectively. The maximum values of the transverse acceleration are therefore

$$\pm\frac{\pi^2N^2r}{10{,}800}.$$

This is the maximum transverse acceleration of the lowest point of the connecting rod. The uppermost point of the rod has no transverse motion, and consequently, no transverse acceleration. The maximum transverse acceleration of any intermediate point at a distance x from the upper end is $x\alpha_m/l$, where l is the center-to-center length of the rod and α_m the maximum transverse acceleration of the lower end,

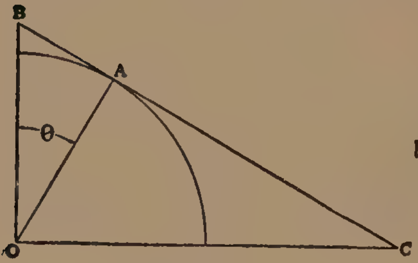


FIG. 163.

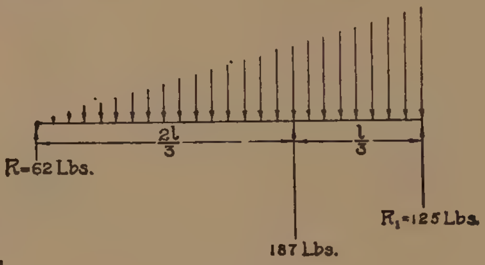


FIG. 164.

Let s represent the cross-sectional area of the rod, which we will consider to be constant from end to end. Then the weight of a small section dx of the rod is

$$W = 0.26\,s\,dx\ lbs.,$$

and the acceleration force on it is

$$\frac{Wa}{g} = \frac{0.26sdx \times \frac{x}{l}\,\frac{\pi^2N^2r}{10{,}800}}{32.16} = \frac{N^2rsxdx}{135{,}320l}.$$

Integrating between the limits $x = 0$ and $x = l$, we have for the total force of acceleration

$$\frac{N^2rsl}{270{,}640}.$$

We will apply this to the example of a connecting rod with a section of the general form shown in Fig. 161, with a cross-sectional area of 0.35 sq .in. and a length of 10 in. between centers. Such a rod might serve for a light truck engine of 4¼-in. bore and 5-in. stroke. To be absolutely safe we must use the maximum speed at which the engine can possibly run, which we may set down at 2400 r. p. m. Then, in the above equation, $N = 2400$, $r = 2.5$, $s = 0.35$ and $l = 10$. Therefore, we get for the force of the transverse acceleration on the rod—

$$\frac{2400 \times 2400 \times 2.5 \times 0.35 \times 10}{270{,}640} = 187 \text{ lbs.}$$

When considered with respect to this force the rod obviously constitutes a simple beam subjected to a non-uniformly distributed load. The sum of the reactions at the supports is equal to the total pressure, viz., 187 lbs. Besides, the two reactions will bear to each other the same relation as if the rod was subjected to a concentrated load at the center of gravity of the distributed load. In Fig. 164 the distributed load is represented by downwardly pointed arrows bearing against the base line, which latter represents the length of the rod. This distributed load can be balanced by a concentrated load of 187 lbs. applied at a point one-third the length of the rod from the lower end, since the distributed load is represented by a triangle, and the center of gravity of a triangle is at one-third its height from the base line. But a reaction of 187 lbs. at one-third the connecting rod length from the lower end is easily resolved into reactions of 125 lbs. at the lower end, and 62 lbs. at the upper end of the rod, as shown in the figure.

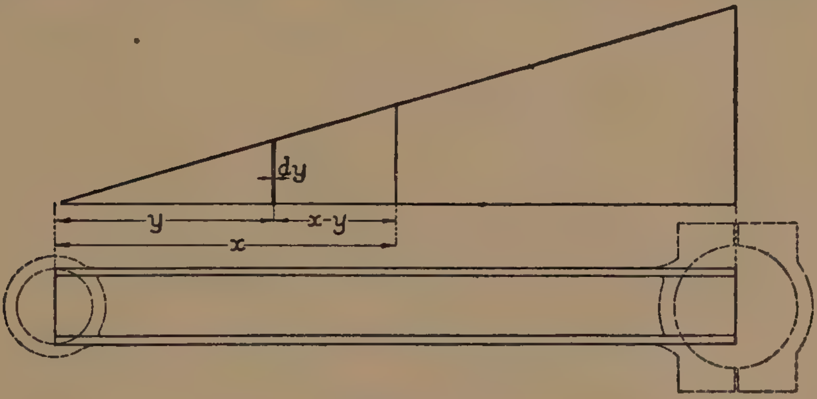


Fig. 165.

We must now find the maximum bending moment on the rod due to this distributed force of acceleration. Referring to Fig. 165, assume any point at a distance x from the upper support. The bending moment at this point is equal to the moment of the reaction at the left-hand support around the point, minus the moment of that part of the distributed force to the left of this point, around the point. The moment of the reaction at the left-hand support is evidently equal to $62x$. In order to find the moment of that portion of the distributed force to the left of the point, we take a section dy to the left of the point x. Since the total length of the rod is 10 inches, the average pressure on it is $187 \div 10 = 18.7$ lbs. per inch in length, and the maximum is twice this, or 37.4 lbs. per inch. The pressure at a distance y from the left-hand support is $37.4y/10$ lbs. per inch in length, and the pressure on a short length dy at that distance from the left support is

$$37.4 \times \frac{y}{10}\,dy = 3.74y\,dy.$$

The moment of this force around a point at a distance x from the left-hand support is

$$3.74y\,dy(x - y) = 3.74(xy\,dy - y^2dy).$$

Integrating between the limits $y = 0$ and $y = x$ we get the expression

$$\frac{3.74}{2}x^3 - \frac{3.74}{3}x^3 = 0.62x^3.$$

Hence the bending moment around a point at a distance x from the left-hand support is

$$M = 62\,x - 0.62\,x^3.$$

In order to find the point where the bending moment is a maximum we place the first differential coefficient of the expression equal to zero—

$$62 - 3 \times 0.62x^2 = 0,$$

from which we find

$$x = 5.77 \text{ inches.}$$

Inserting this value of x in the equation for the bending moment we have

$$M = (62 \times 5.77) - (0.62 \times 5.77^3) = 238.6 \text{ lb.-in.}$$

From Chart III we find the thickness t of a section proper for a connecting rod 10 inches long and subjected to an explosion pressure of 5300 lbs. if the stress is to be 18,000 lbs.

per sq. in. to be 0.185 inch, and since the moment of inertia around the principal axis perpendicular to the plane of motion is

$$I_x = 46.8t^4,$$

its value in this case is

$$46.8 \times 0.185^4 = 0.052.$$

The distance c of the outermost fiber from the neutral section is

$$\frac{5.7 \times 0.185}{2} = 0.527 \text{ inch.}$$

Hence, since the stress

$$S = \frac{Mc}{I},$$

$$S = \frac{238.6 \times 0.527}{0.052} = 2400 \text{ lbs. per sq. in. (appr.)}$$

This stress must, of course, be added to that due to the "static load" on the rod. When the crank during the power stroke is in the quarter position, the piston is 2.8 inches from the top end of the stroke (see Fig. 14), and the gas pressure in the cylinder then is 65 lbs. per sq. in. (see Fig. 16). The inertia force due to the reciprocating parts is *nil* at this point.

The section of the rod has been calculated so that with a gas pressure of 400 lbs. per sq. in. the maximum stress in the rod is 18,000 lbs. per sq. in. Hence with a gas pressure of 65 lbs. the stress is

$$\tfrac{65}{400} \times 18{,}000 = 2925 \text{ lbs. per sq. in.}$$

The combined stress therefore is

$$2400 + 2925 = 5325 \text{ lbs per sq. in.}$$

which is very low compared with the 18,000 lbs. per sq. in. to which the rod is subjected at the moment of explosion. This calculation shows that with steel I-section connecting rods there is absolutely no danger from the whipping effect, even at the very highest speeds.

Piston Pin Bearing—When the piston pin is clamped in the connecting rod it usually bears directly on the metal of the piston bosses, as already pointed out. Lubrication is then provided for by drilling inclined oil holes from the oil groove below the ring groove just above the pin, through ribs on top of the bosses.

The boss at the upper end of the connecting rod is made from 1/8 to 3/16 in. narrower than the space between the piston bosses, so that in spite of a certain amount of axial play of the crankshaft there may be no cramping of the piston. Of course, the longer the engine the greater the chances of inaccuracy in alignment, and with four-cylinder engines the axial play of the connecting rod small end can be limited to the lower value given.

Methods of fastening the piston pin in the connecting rod have been described already. It only remains to say that the clamp screw must be of substantial size and securely locked, and that there must be sufficient metal around the clamp screw (particularly if it passes through the shank) so the rod will not be weakened.

For passenger-car engines the bronze bushing for the top end of the connecting rod is made either 7/64 or 1/8 in. thick and of exactly the same length as the hub. It is pressed into the latter and as a rule is held in place merely by friction, though a few makers secure it by a pin serving as a key. The hub, of course, tapers from the middle toward both ends, partly because draft is required for forging and partly because the stresses are greatest at the middle, that being where the forces are transferred to the shank. At the ends the wall thickness may be made ⅛ in., and the usual 7° taper then gives an adequate section at the middle.

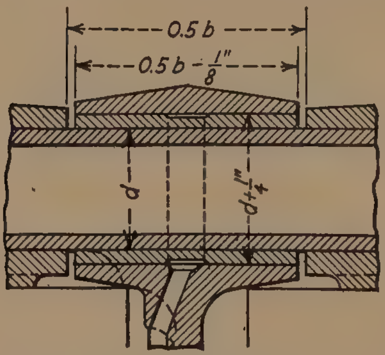


FIG. 166.—CONNECTING ROD SMALL END.

In a few engines oil is carried to the piston pin bearing under pressure through a tube extending up the shank, as indicated in Fig. 166. In general the top end of the rod is supplied with lubricant by drilling a hole through the top of the boss and counter-sinking it so it will catch more of the oil spray. To obviate the possibility of the oil passage being obstructed by an angular displacement of the bushing, the latter has an oil groove turned in its outer surface, in line with the oil hole through the boss, at the bottom of which a number of holes are drilled through the wall of the bushing. Sometimes oil grooves are turned also on the inside of the bushing, so as to carry the oil from the entrance, around and toward the ends of the bearing.

Connecting Rod Head—The connecting rod head is made in halves, with a detachable cap held on by two bolts in the smaller sizes and by four in the case of rods for large truck engines. Splitting the head, of course, is necessary in order to get it over the crankpin. At the head end, bearings of white metal or babbitt are used, because they do not wear the crankshaft as much as the harder bronze bearings, and the lower unit pressures and lower temperatures permit of their use.

An older practice, which is still followed to some extent in truck engines, is to fit the connecting rod head with two half bushings, one secured to the rod and the other to the cap, by anchoring screws. The bushings are sometimes cast entirely of white metal (generally by the die-casting process) but more frequently the white metal forms merely a facing for a bronze shell, white metal having the more desirable bearing qualities and the bronze superior strength.

With this form of construction it is usual to place a number of thin steel or copper shims of varying thickness between the rod and the cap, of which one can be removed after the bearing has worn and requires adjustment.

At present the babbitt in most cases is cast directly into the steel rod, and shims are generally dispensed with. One advantage of the newer practice is that it permits of a thinner layer of bearing metal, which results in a reduction of the centrifugal force; and another that there is a better contact between the bearing metal and the steel of the rod, consequently the heat generated is carried off more readily and the bearing remains cooler. When the big end bearing has worn so much that it has become noisy, the whole rod is replaced.

The length and inside diameter of the connecting rod head are fixed by the dimensions of the crankpin and need not be discussed here. If a removable bushing is used its thickness can be made $d/12$, where d is the crankpin diameter; while if the babbitt is cast directly into the rod the layer can be made quite thin, from 1/16 to 3/32 in.

Designs of connecting rod head are shown in Figs. 167-A and 167-B. The first of these shows an older design for a truck engine. It is of the four-bolt type, with half-bushings and shims. The method of fastening the half-bushings by means of flat-head screws, to the rod and cap respectively, is indicated. The bushings are relieved at the sides, and as the oil is fed to the bearings through holes drilled in the crankshaft, these reliefs are always filled with oil. From the relief shown in the drawing there lead two oil grooves toward the left end of the bearing, and from the other relief two similar grooves lead toward the right end.

At B is shown a head end in which the babbitt is cast in and which is without shims. This connecting rod is also intended for an engine with pressure lubrication.

Where splash lubrication is still being used, as it is on some of the cheaper passenger-car engines, the cap is often formed with an oil scoop, or dipper. This dips into the oil trough and splashes oil over all the interior parts, and it also scoops it up and conducts it to the crankpin bearing, for which purpose a hole is drilled through it and the wall of the cap.

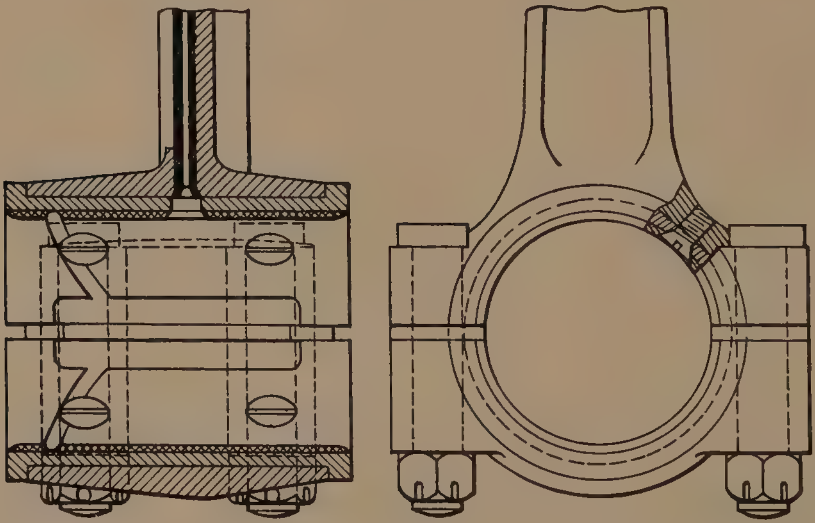


FIG. 167-A.—CONNECTING ROD HEAD WITH REMOVABLE BEARINGS.

The heads have the caps forged integral with them and the caps are later sawed off. This facilitates the operation of boring out. The heads are given the usual drop forging draft and are provided with bosses for the cap bolts.

Form and Size of Cap Bolts—Connecting rod cap bolts have been standardized by the Society of Automotive Engineers, and a drawing of the bolt and a table of sizes are given in the Appendix. The method of calculating the bolt diameter required may be illustrated by an example. We will assume that the 3½ by 5-in. engine which have been used repeatedly as an example, is capable of a maximum speed of 3000 r.p.m. The greatest strain comes on the bolt while the engine is running at maximum speed, when the piston starts on the suction stroke, at which point of the cycle the resultant force in the direction of the cylinder axis is a maximum. Since the maximum inertia force on the piston of this engine at 1800 r.p.m. is 129.5 lbs. per sq. in. (See Table II) and since the inertia force varies as the square of the speed, at this high speed the force is

$$9.62 \times 129.5 \times \frac{3000^2}{1800^2} = 3600 \text{ lbs.}$$

In calculating this force, only one half of the connecting rod weight is taken into account, the other half being considered to have a rotary motion. This half, however, is subjected to centrifugal force and at the beginning of the intake stroke this force is in the same direction as the inertia force of the reciprocating parts. Of course, the centrifugal force on the cap is transferred directly to the crankpin and need not be considered here. That portion of the rod proper which has not been included in the reciprocating weight may be taken to weigh 1.5 lbs. The centrifugal force which produces stress on the cap bolts then is

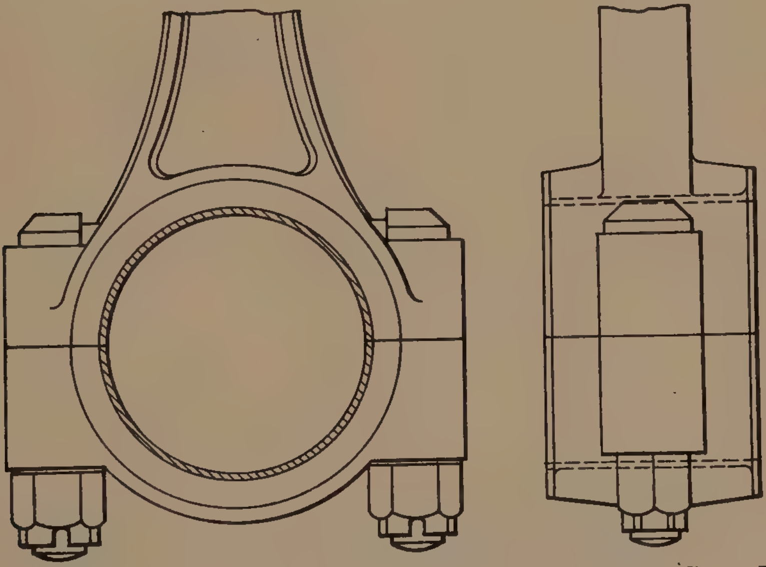


FIG. 167-B.—CONNECTING ROD HEAD WITH BABBITT CAST IN.

$$1.226 \times 1.5 \times \left(\frac{3000}{60}\right)^2 \times \frac{2.5}{12} = 956 \text{ lbs.}$$

The total force on the cap bolts therefore is

$$3600 + 956 = 4556 \text{ lbs.}$$

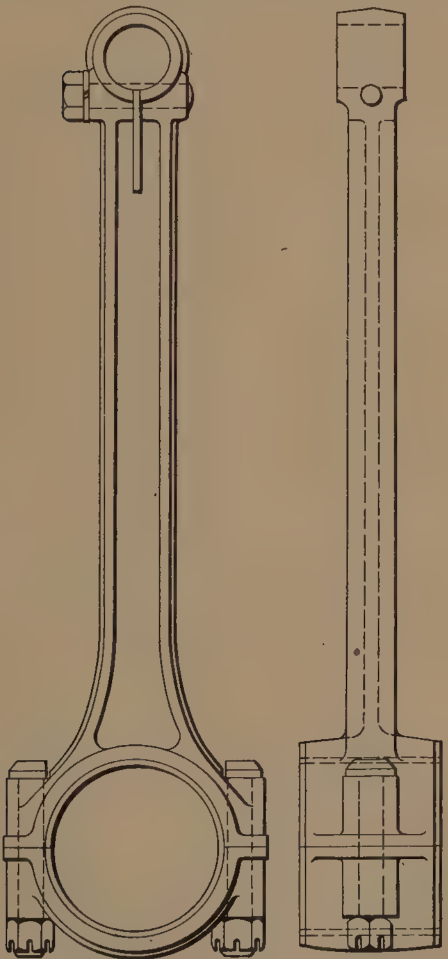
Figuring on a stress of 20,000 lbs. per. sq. in. and remembering that the load is taken on two bolts, the cross-sectional area of each bolt must be

$$4556/(2 \times 20{,}000) = 0.114 \text{ sq. in.}$$

This is substantially the area of a 7/16-in. S. A. E. screw at the bottom of the thread, and screws of this size should be used. In as much as the load is a dynamic one, the stress of 20,000 lbs. per sq. in. is rather high, and it is therefore advisable to use nickel steel bolts.

With connecting rod heads 2½ inches or more in length it is evidently a good plan to use four instead of two bolts, as there is then less danger of the bearing surface being distorted. The only objection to four bolts is that it is impossible to forge the rods with two bosses on each side. One wide boss must be forged on and some of the material at the middle milled away so as to form two bosses. In this connection it is worth pointing out that a favorite practice with European manufacturers is to mill away the central portion of each cap bolt boss, which is dead weight and adds unnecessarily to the load on the crankpin.

Offset Connecting Rods—With the object of making their engine as compact as possible without skimping the bearing surfaces, some designers offset the connecting rods, as shown on an exaggerated scale in Fig. 170. The following consideration shows that very little, if anything, is gained by this procedure. The resultant of the distributed pressure on the piston pin bearing must be equal to and in line with the resultant of the pressure at the crankpin bearing. Let us consider the two bearings equally offset from the rod in opposite directions. Then the resultants of the pressures at the bearing surfaces will be represented by *R*. These forces, it will be noted, are to one side of the centers of the bearings, and we found in the discussion of the side thrust of pistons that when a bearing surface is thus unsymmetrically loaded the pressure is unevenly distributed over the surface. The unit pressure at the end farthest away from the point where the resultant takes effect is to the unit pressure at the opposite end as the distance of the nearest end from the resultant is to the distance of the farther end. Suppose, for instance, that in Fig. 170 the two bearings are of equal length, and that the two "overhanging" portions of the bearings are equal to one-third the length of the bearing each; then the location of the resultant will be one-third the length from one end of the bearing and the unit pressure will be twice as great at the nearer end as at the farther end. Calling the average unit pressure p, the unit pressure at the nearer end will be $4p/3$. The wear at the near end will, of course, be proportional to $4p/3$, or 33 per cent greater than if the bearings were of the same size and in line. Now, if the overhanging portions were left off the unit pressure would be uniform, $3p/2$, throughout the length of the bearing, or only about 13 per cent greater than with this big offset. In an offset rod the bearings will wear more rapidly at one end than at the other, and will then subject the connecting rod to additional bending strain.



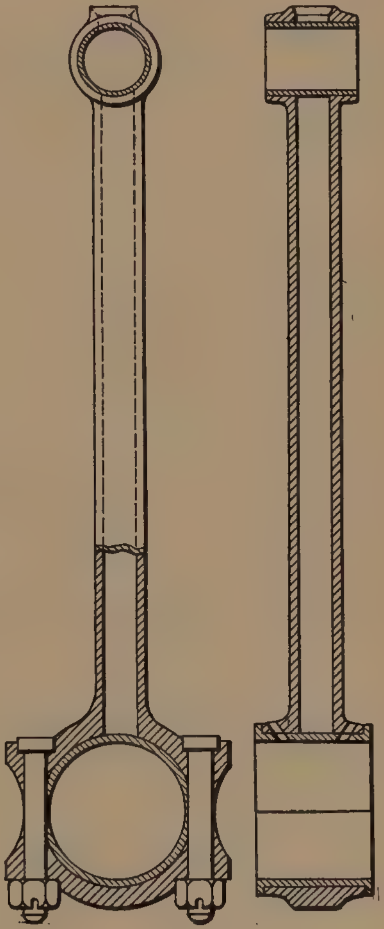


FIG. 168. (LEFT).—TYPICAL I-SECTION CONNECTING ROD.
FIG. 169 (RIGHT).—TUBULAR CONNECTING ROD.

Forked Connecting Rods for V Engines—In all V engines the connecting rods of two oppositely located cylinders connect to the same crankpin. There are three possible arrangements, as explained in the chapter on Crankshafts, but the forked arrangement is the one in most common use. One objection to the forked connecting rod as ordinarily constructed is that it cannot be taken up for wear. Referring to Fig. 171, which illustrates this type of rod, the bearing bushing is made in halves, and the end of the forked rod is securely clamped to the bushing. The other connecting rod has its bearing on the bushing between the two prongs of the forked rod. It will be readily seen that if the bearing were adjusted to compensate for wear at the crankpin surface, the outside of the bushing would be too small to fit the bearing of the plain connecting rod. Fortunately, the pressures on the individual connecting rods in a V engine are relatively small, and it is possible to keep down the unit pressures on the crankpin bearing to a low figure. There is very little wear on the bearing of the plain rod, as the bushing does not revolve in this bearing, but merely rocks back and forth in it through a small angle.

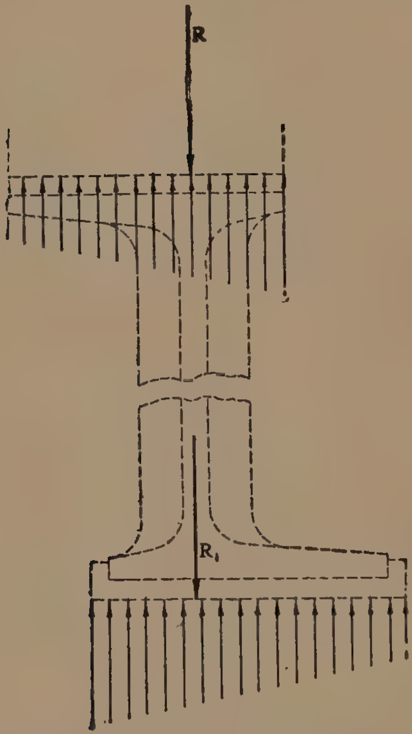


FIG. 170. — UNSYMMETRICAL CONNECTING ROD.

Duralumin—Of late years light alloy connecting rods have come into use to a certain extent. The alloys principally used belong to a class known as duralumin. These are forgeable alloys of aluminum, copper and magnesium, the copper content ranging between 3.5 and 4.5 per cent, the magnesium content between 0.2 and 0.75 per cent, and the manganese content between 0.4 and 1.00 per cent, while some specifications call for a minimum aluminum content of 92 per cent.

Duralumin was originally patented in 1903 by A. Wilm, a German metallurgist, and has been widely used for the framework of Zeppelins and the fuselages and other parts of metal airplanes. In the United States two "brands" are being manufactured at the present time, known, respectively, as Lynite S-17 and Huron metal.

What makes duralumin more valuable than other light alloys for connecting rods and similar highly stressed parts, is that, like steel, it is responsive to heat treatment, and that its mechanical properties can be greatly improved by such treatment. When properly treated, duralumin has about the same strength as mild carbon steel. Although its specific gravity is only 2.82, it has a tensile strength, in the treated state, ranging from 55,000 to 60,000 lbs. per sq. in.; an elastic limit better than 30,000 lbs. per sq. in. and an elongation of 18-22 per cent in 2 in. The Brinell hardness varies between 94 and 100.

The action of heat treatment is quite different upon duralumin than upon steel, for whereas the improvement in mechanical qualities appears in the steel immediately after quenching from a temperature above its critical point, duralumin is comparatively soft immediately after quenching, and the quenching must be followed by an "aging" process. The heat treatment usually applied consists in raising to a temperature of 920°-950° F., holding at this temperature for 20 minutes or longer, until the metal is thoroughly saturated, and quenching in water (either hot or cold). Then the metal is allowed to age for from four to ten days at normal atmospheric temperature. During this period it reaches its maximum strength, and thereafter there is no change in the properties, provided the alloy is not exposed to excessive temperatures.

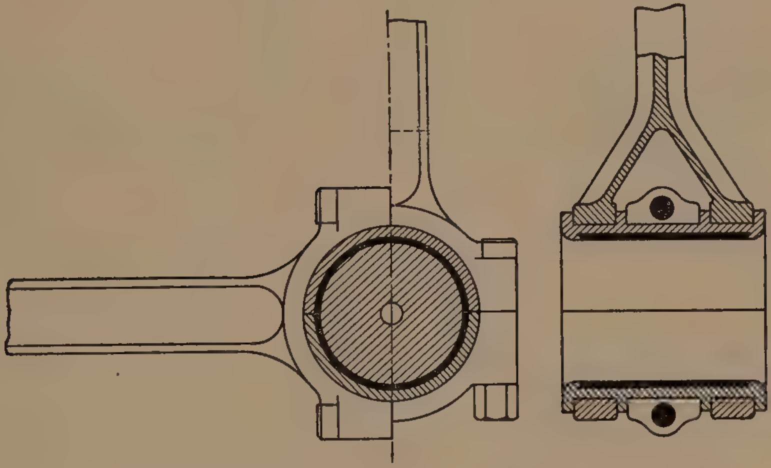


FIG. 171.—YOKED CONNECTING RODS FOR V ENGINE.

Heating for quenching is generally accomplished in a salt bath composed of equal parts of sodium nitrate and potassium nitrate. When at the quenching temperature the alloy is quite soft, and must therefore be well supported and carefully handled. Duralumin forms a good bearing surface for hardened steel.

Stress Permissible in Duralumin Rods—The cross-section of the duralumin connecting rod used on the Hupp vertical eight is shown in twice natural size in Fig. 172. The height of the section is 7/8 in. and the width 5/8 in., and as near as can be figured, the cross-sectional area is 0.394 sq. in. and the moment of inertia around the transverse axis, 0.0307. Hence the square of the least radius of gyration around this axis is 0.078. This engine has a cylinder bore of 2 7/8 in. and therefore a cylinder head area of 6.49 sq. in. Figuring on a maximum explosion pressure of 400 lbs. per sq. in., the total force of explosion on the piston would be 2600 lbs. This would give a stress in the material (equation 62) of

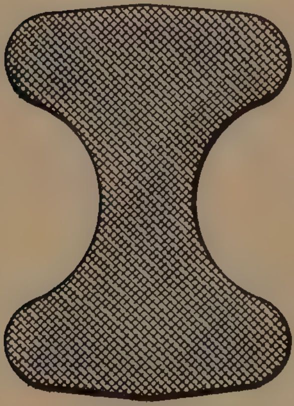


FIG. 172.—CROSS-SECTION OF DURALUMIN CONNECTING ROD.

$$\frac{2600}{0.394} + \left(0.000526 \times \frac{2600}{0.394} \times \frac{9.5 \times 9.5}{0.078}\right)$$

$$= 6600 + 0.000526 \times 6600 \times 1157$$

$$= 6600 + 4000 = 10{,}600 \text{ lbs. per. sq. in.}$$

Considering that the tensile strength of duralumin is about 60,000 lbs. per sq. in., as compared with 95,000 lbs. for 0.35 per cent carbon steel, heat-treated, this corresponds well with the 18,000 lbs. per sq. in. recommended for carbon steel connecting rods.

Rods of Other Light Alloys.—Some use has been made also of other aluminum alloys, notably one of aluminum and silicon, known as Alpax or Silumin. This usually contains about 10 per cent of silicon, and some other ingredients added to improve its machining qualities. Although Alpax is being manufactured in America as well as in Europe, it has been used for this purpose more abroad. In 1925 the Omnibus Department of the Paris Public Service Company ran tests on pistons and connecting rods of Alpax and also on pistons of Alpax used in conjunction with connecting rods of magnesium. In bench tests the pistons and connecting rods of Alpax, when used in an engine operating with a compression ratio of 5.4:1 (made possible by the use of non-detonating fuel) resulted in a saving of 3.5 per cent on the fuel consumption, but in road tests with ten omnibuses this gain in economy fell to 1 per cent.

The magnesium rods, although made of larger cross-section, were unable to stand the strain, and all of them broke after comparatively small mileages. These data are taken from a report on the use of light alloys in omnibuses, made at the 1925 convention of the Union of Railway and Motor Services, by M. A. Banlier, Chief Engineer of Bus Equipment, Paris Public Service Company.

One of the desirable qualities of aluminum-silicon alloys is that their specific gravity is lower than that of aluminum, whereas all of the common commercial alloys, containing zinc or copper, have a higher specific gravity. The specific gravity of silicon is only 2.42, as compared with 2.70 for aluminum. Another advantage is that the heat expansion, and, therefore, the necessary shrinkage allowance in patterns, is less than with other aluminum alloys.

In considering the advantages of light alloys for connecting rods, one factor that needs to be taken into account is the effect of the necessarily larger section on the stiffness of the rod. For instance, if two rods were made, of steel and duralumin respectively, with sections similar in shape and of areas inversely proportional to the respective tensile strengths, then the duralumin rod would have the greatest factor of safety, because the least radius of gyration of its section would be materially greater. However, it appears that in practice this gain is not realized, because, except where the duralumin rods are forged in dies originally made for steel, the sections, that is, the thicknesses of the web and flanges, are made relatively heavier.

Manufacture of Connecting Rods—The first machining operation on connecting rod forgings generally consists in rough-boring the holes at opposite ends, which is done in a double spindle drill press, with the rod held in a fixture. Sometimes, however, when the ends of the two bosses are finished off by straddle milling, this operation comes first. If the boring out is done first the ends are usually finished by means of face millers carried on fluted pilots passing through the rough-bored holes and each driving an additional face miller working on the lower face. The spindles are fed down until they come up against a stop, when the hubs will be reduced to the proper width, with faces substantially at right angles to the axes of the bores. This operation is performed in a double spindle machine of the same type as that used for the boring operation.

Next the piston pin hole may be finish-reamed and broached. The cap bolt bosses are finished off at both ends by means of a straddle miller. This operation is located from the two bored holes, and the rod is clamped at the faces of the big end. After this comes the drilling and tapping of the bolt holes. This is usually done in multiple spindle drills, and the method of doing it in the Hupp Motor Car Co. plant is illustrated in Fig. 173. A three-station fixture is used, with the stations 120° apart. At one station the holes are drilled and at another they are tapped, while the third station serves for reloading. For greater accuracy the bolt holes are often reamed.

The caps are parted from the rods by means of a saw in a milling machine, and this operation may be combined with a finishing operation on the ends of the cap bolt bosses, three rotary cutters being used.

Babbitting the Big Ends—One of the most interesting operations on a connecting rod is that of babbitting the big end. This is now generally done either by the die casting or the centrifugal casting process. In order that the babbitt may get a firmer hold, the bored surface is left quite rough, and as a further aid in securing a firm bond the surface of the steel is first tinned. To prevent the tin from adhering to near-by finished surfaces where it is not wanted, such as the ends of the cap bolt bosses, these are first given a coat of whitewash.

For the tinning operation the cap and rod—the former held in tongs gripping it at the bolt holes—are dipped first into a preheating bath. Next they are covered with flux at the surface to be coated, either by hand by means of a brush or by a slowly revolving felt wheel partly immersed in the flux. Finally they are immersed in the bath of molten tin and agitated while they rise to the temperature of the tin. Such baths of white metal are generally heated electrically, their temperature being controlled automatically.

The babbitting process proper differs according to whether the die casting or centrifugal method is followed. Some makers use the Chadwick-Leclair die casting machine. With this the big end is held in a die and the molten metal is forced into the die by a hand pump which is submerged in the pot. The die is operated by compressed air, which the operator controls with his foot, so that he has both hands free to work the pump and for other necessary manipulations. The permanent parts of the die are water-cooled. After the rods and caps have cooled the gates are removed, the joint surfaces cleaned by filing, and the caps are bolted on.

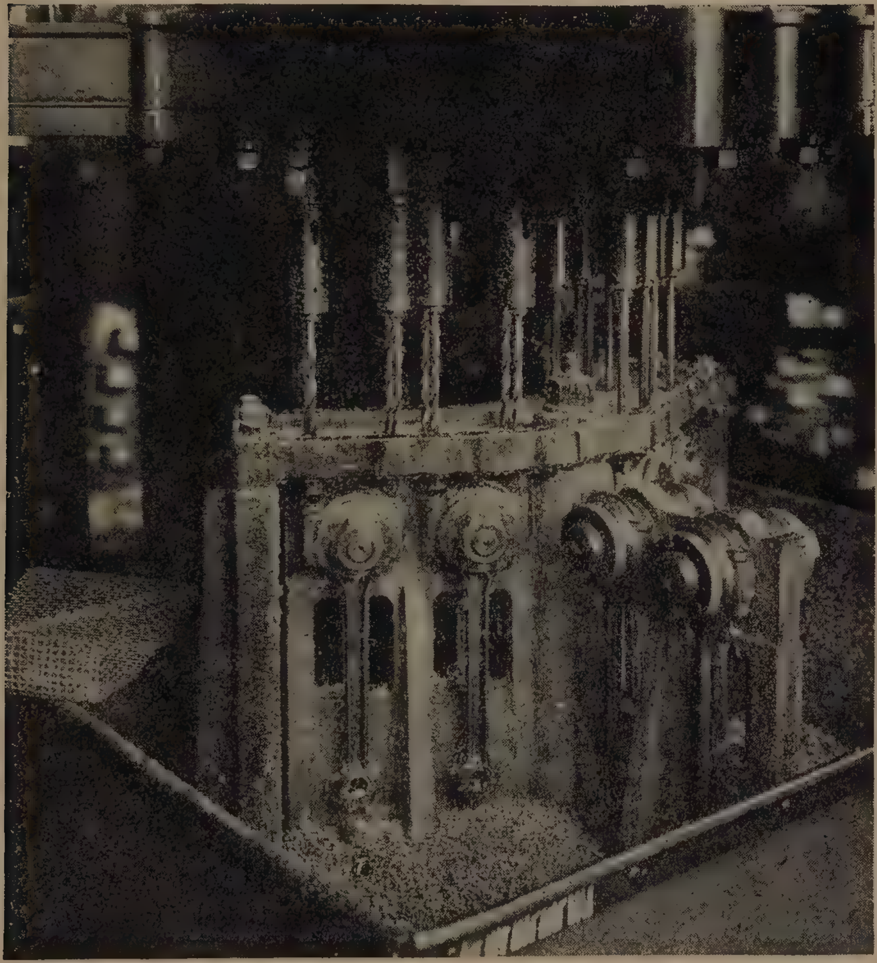


FIG. 173.—DRILLING CAP-BOLT HOLES.

If the centrifugal method is used, the rod is clamped to a spindle which is driven from a line shaft or an electric motor at a speed of perhaps 800 r.p.m. The head end is clamped against a face plate, and a plate with a central opening in it is clamped against the outer side. Then, while the rod is revolving, the babbitt is poured into the head

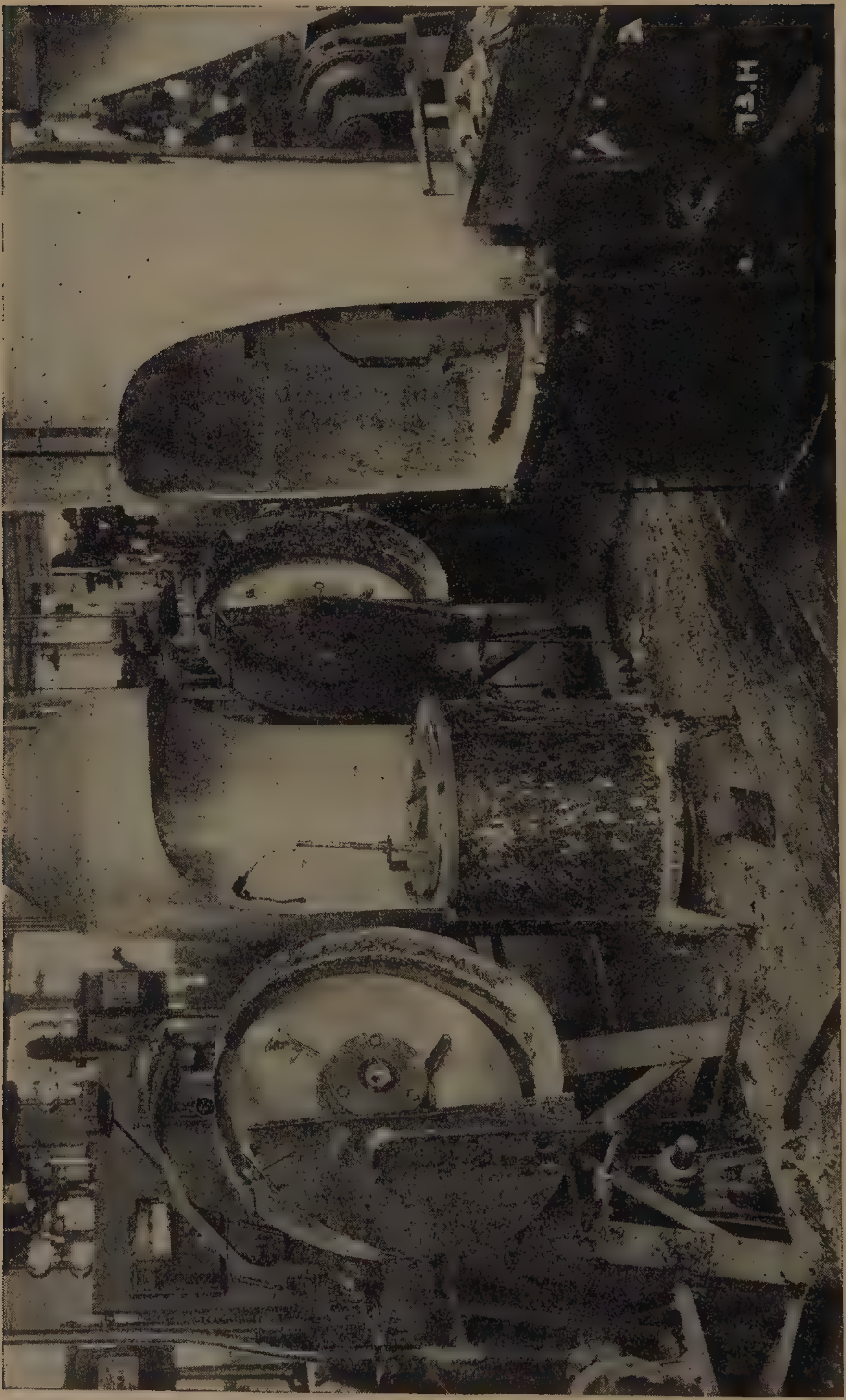


Fig. 174.—Installation for Babbitting Connecting Rod Heads "Centrifugally" (Hupmobile).

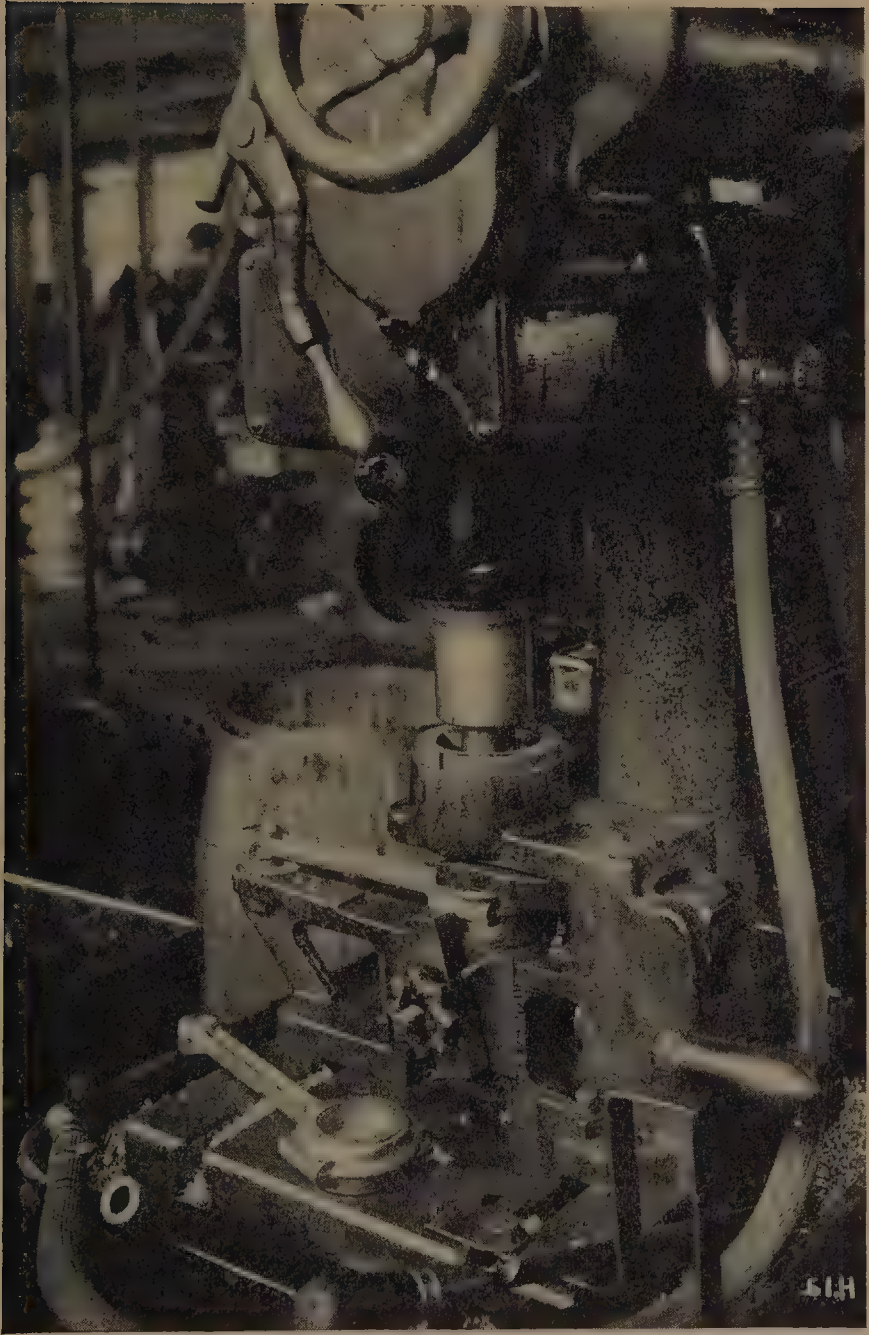


FIG. 175.—ROUGH-BORING CONNECTING ROD HEAD (HUPMOBILE).

with a ladle, and owing to the centrifugal force on it, it covers the bore uniformly. A connecting rod babbitting installation with its melting pots and centrifugal fixtures is shown in Fig. 174. This is from the Hupp Motor Co. plant. Owing to the pressure on the babbitt due either to centrifugal force or the pump, it develops a relatively dense structure.

In some cases, in order to reduce the amount of babbitt required and obtain a lining of uniform thickness all around, the big end is "ovaled" in a press before the head is detached, so that the hole will be cylindrical after that operation.

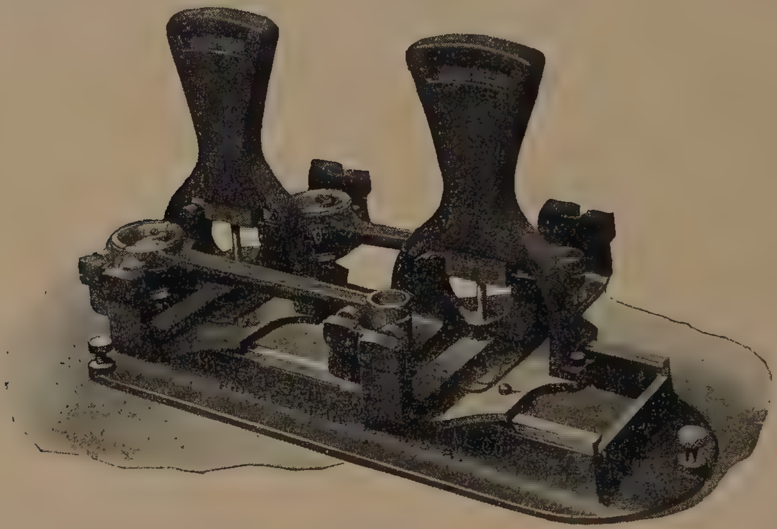


Fig. 176.—Olsen Special Balancing Scales.

Finishing Big End—After the babbitting comes the boring and finishing of the big end. According to M. R. Wells, this is done as follows in the plant of the Cleveland Automobile Co.: Rough-boring is done in a double-end boring machine, the rod being located for the operation by clamping a piston pin in it and holding this pin between V blocks at the proper distance from the big end. The big end is clamped on the bolt bosses between two V blocks. One boring head feeds in to rough-bore, face off and chamfer one end of the head; then it withdraws and the other feeds in from the opposite side, reams, faces and chamfers the other end.

The head end or crankpin bearing is mostly finished by means of a combination broach and burnishing tool. The broach removes some metal, producing a very nearly true cylindrical hole of standard size, and the burnishing portion of the tool then slightly enlarges this by compacting the babbitt. Owing to the lesser rigidity of the boss in the transverse direction, the transverse diameter of the hole will be slightly greater than the diameter in line with the shank, but this is considered an advantage, as it gives a slight relief on the sides, which tends to lessen bearing friction.

Inspection—The completed rod with the small end bushing in place is inspected for parallelism and squareness of the two holes, and is then weighed, a frequent practice being to pick rods of closely similar weight for use together in the same engine (selective assembly). Some manufacturers are not content with making sure that all of the rods of a set weigh the same within a certain tolerance, but check the distribution of weight between the two ends. This is done by means of a special double scale as shown in Fig. 176. Before being assembled the rods are cleaned of grease and grit by means of some cleaning solution, such as sodium phosphate and soft soap.

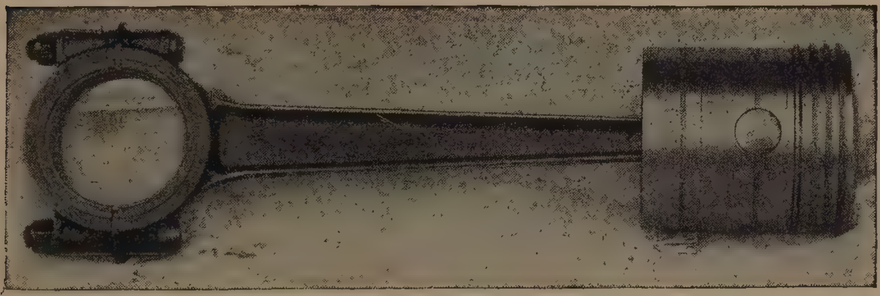


Fig. 177.—Assembled Piston and Connecting Rod.

If separate bushings are used at the big end the procedure in production necessarily follows somewhat different lines. The hole at the big end is sometimes ground out to insure as good a contact as possible between the steel and the liner. Lock screw holes must be drilled and tapped in both the rod and the cap, and after the bushing halves are put in place they are face-ground flush with the rod and cap. The bushing at the small end may also be finished by a burnishing operation or one similar to it in its effects.

CHAPTER XII

Valves and Valve Gears

The modern standard automobile engine works on the four-stroke principle, as explained in the chapter on "The Otto Cycle," and is fitted with mechanically operated poppet valves for both the inlet and the exhaust. A poppet valve consists of a disc of metal, with a coaxial stem on one side, which closes a circular opening in a partition wall between two chambers, against which wall it is drawn by a spring. To open the valve a force has to be applied to it contrary to the direction of spring pressure, strong enough to overcome the spring. In the early engines the inlet valves were actuated by the suction prevailing in the cylinder during the inlet stroke, such valves being called automatic or suction valves. A typical valve of this kind with its cage is illustrated in Fig. 178. It will be understood that, since the valve opens toward the cylinder, when the piston moves down on the inlet stroke and the pressure in the cylinder drops below atmospheric, the excess pressure of the atmosphere against the outer face of the valve head overcomes the pressure of the light valve spring, and the valve opens. Toward the end of the inlet stroke the excess pressure of the atmosphere diminishes and the spring then closes the valve. Automatic inlet valves are no longer used and are of historical interest only; they are not suited to high speed engines.

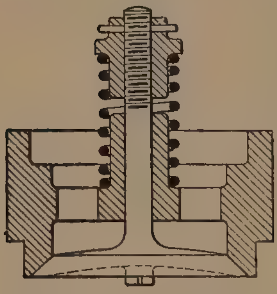


FIG. 178.—AUTOMATIC INLET VALVE WITH CAGE.

Angle of Seat—The poppet valve consists of two parts, the head and the stem. That portion of the head which when the valve is closed rests on a machined portion of the cylinder casting is known as the seat. This surface usually forms a truncated cone whose generatrices make an angle of 45° with the plane of the valve head. Sometimes, however, the valve seat is a plane surface perpendicular to the valve axis, and, again, the seat is a truncated conical surface of less than 45° inclination. Flat-seated valves have been used by some prominent makers, but at present the great majority use 45° conical valves. The advantage of the flat-seated valve is that for a certain area of opening at the seat it does not need to be lifted as high as a conical valve. It can, therefore, attain its maximum opening quicker and retain it longer, and since the amount of travel in closing is less, it should operate more quietly. The conical valve, on the other hand, has the advantage that it is self-centering. The two types are shown in Figs. 179 and 180, respectively.

In Fig. 179 let d ($= B\,B'$) be the clear diameter of the valve or the diameter of the bore of the valve seat, and h ($= B\,D$) be the lift. Also, let θ be the angle of the valve seat. It is required to find the effective area of opening corresponding to any lift h. Referring to the figure, the minimum area of cross-section is evidently a truncated conical surface, of which $B\,C$ is a generatrix. Now,

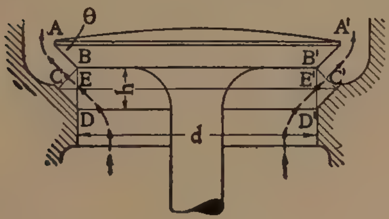


Fig. 179.—Conical Valve.

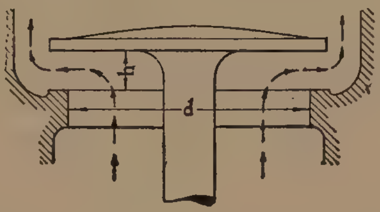


Fig. 180.—Flat-Seated Valve.

$$BC = BD \cos CBD = h \cos \theta.$$

Also

$$CE = BC \cos BCE = BC \sin BCE = h \cos \theta \sin \theta.$$

Hence the diameter CC' is

$$d + 2h \cos \theta \sin \theta.$$

The area of the truncated conical surface is

$$\pi \frac{CC' + BB'}{2} \times BC = \pi(d + h \cos \theta \sin \theta) h \cos \theta$$

$$= \pi(dh \cos \theta + h^2 \cos^2 \theta \sin \theta). \qquad (66)$$

In case $\theta = 45°$ the expression for the area becomes

$$A = \pi(0.707dh + 0.353h^2). \quad . \quad . \quad . \quad . \qquad (67)$$

It will readily be seen by reference to Fig. 180 that in the case of a flat-seated valve, the area of opening is πdh. Hence, calling the lift of the flat valve h_f and that of the 45° valve h_{45} the two lifts must be so related that

$$0.707dh_{45} + 0.353h^2{}_{45} = dh_f,$$

from which it follows that

$$h_f = 0.707h_{45} + 0.353\frac{h^2{}_{45}}{d}.$$

If the lift of the 45° valve is one-fifth the clear diameter, then

$$h_f = 0.778h_{45},$$

or roughly four-fifths of h_{45}.

It would be desirable in one respect to have the passage around the valve when fully lifted about equal to the area of the valve port. The full area of the valve port is $0.7854d^2$, but the stem will reduce this to $0.74d^2$. Hence

$$3.14(0.707 h + 0.353h^2) = 0.74d^2,$$

$$2.22dh + 1.11h = 0.74d^2,$$

which when solved gives

$$h = 0.29\ d.$$

In practice, however, the lift is always made less than this, owing to the difficulty of making the valve action sufficiently quiet with high lifts. There is also a noticeable tendency to make the lift of small valves proportionately greater than that of large valves. A good rule is to make the lift

$h = (d/8) + 1/8$ in. for passenger-car engines

and

$h = (d/8) + 1/4$ in. for larger truck engines.

Valve Operating Conditions—The heads of the valves are subjected to the high temperature of the burning gases, and it is essential that they should not warp under the influence of the heat, and that their seats should not scale or oxidize, as in either case they would become leaky. Occasionally small particles of scale will get onto the valve seats, and the valve heads must be of sufficient hardness at the high temperature at which they operate so they will not pit under this condition.

Lubrication of the valve stems is hard to effect, and the stems must not wear too rapidly in their guides even though poorly lubricated or not lubricated at all. That portion of the stem immediately below the head is also subject to the heat of the burning gases, which, when the exhaust opens, rush by it at a velocity of about 200 ft. per second under conditions of maximum load; and to the corrosive action of unconsumed oxygen and compounds formed by any sulphur that may be present in the fuel.

Finally, the tip of the stem receives a quick succession of blows from the tappet as the clearance is being taken up, and it must be sufficiently hard to withstand these blows without undue wear. Hardening of the tips is usually effected by the so-called cyaniding process (dipping in a bath of potassium ferro-cyanide).

A rather extensive investigation of the temperature conditions of valve heads in automotive engines was made by Prof. A. H. Gibson and H. W. Baker of the University of Manchester, England, who presented their results in a paper read before the Institution of Mechanical Engineers, December 14, 1923. Tests were made on large bore, air-cooled, aircraft engines, as well as on a 3½ by 5¼ in. single cylinder unit made up to the same design as the cylinders of an Armstrong-Siddeley six-cylinder, 20 h.p. water -cooled automobile engine.

The object of the tests was to determine how the valve head temperature varies with different factors, so as to discover ways and means of lowering this temperature where the valves will not stand up. The results with valves in the water-cooled unit are of particular interest here.

Valve Temperature Tests—Measurements of temperature were made by two different methods, viz., by means of a thermocouple inserted in the valve head and by a radiation pyrometer, the latter method being made possible by running on an "open exhaust" so that the pyrometer could be aimed at the head of the valve through the exhaust port. The results of the two measurements checked remarkably well.

The temperature of the valve rises and falls in a straight-line relation with the horse power output. In the water-cooled 3½ × 5¼ in. unit the exhaust valve reached a temperature of 1364° F. at full load of 10.5 h.p., and down to about half load its temperature fell substantially 35° per horse power.

As far as the effect of mixture ratio is concerned, the maximum valve temperature is attained with a ratio of 15.5 : 1. With a ratio of 11 : 1 it appears to be about 90° F. lower under full throttle and normal full-load speed. The temperature of the valve also depends to a considerable extent upon the engine speed. In the water-cooled cylinder, for instance, with a compression ratio of 4.77 : 1, the exhaust valve temperature dropped from 1400° F. at 1800 r.p.m. to 1220° at 1500 r.p.m. The valve temperature does not appear to be materially influenced by the compression ratio. It decreased as the spark advance was increased. At 1800 r.p.m. it dropped from 1436° F. for 25° advance to 1364° F. at 45° advance.

A reduction in valve temperature of 90° was effected by shifting the spark plug to a different position, and Messrs. Gibson and Baker conclude that it is inadvisable to fit a spark plug either directly adjacent to, or directly opposite an exhaust valve. Owing to the compression during the progress of the explosion, of the gases in the vicinity of the plug, which have already had their temperature raised by the combustion, the gas temperatures near this point are higher than in any other part of the cylinder. The highest exhaust valve temperature measured was 1475°.

If an exhaust valve becomes leaky for any reason, as, for instance, through "dishing" of its head by reason of loss of strength at high temperature, through improper adjustment or through excessive warping, the head will be destroyed very quickly, as the burning gases at the temperature of the explosion (about 3000° at full load) will then blow by it.

Materials for Valves—As a rule both sets of valves are made of the same material, for the sake of interchangeability, but sometimes, on account of the expensive character of the material considered desirable for the exhaust valves, the inlets are made of more common and cheaper material.

Cast iron is a fairly good material for the valves of engines intended for light duty, that is, which seldom run under full throttle and at high speed for any extended period. The head is screwed and riveted, or merely riveted, to a steel stem. One objection to this is that the path for heat flow from the head to the stem is rather poor, and that the cooling facilities of the head therefore are not the best. This it has been sought to overcome recently by welding the heads to the stems.

Ordinary carbon steel, drop forged valves are also being used, but are not suitable for heavy duty service. The first substitute for carbon steel to come into use was a 3 per cent nickel steel, and later a high nickel steel of about 36 per cent nickel was tried, but the trouble with such materials is that the stem is not sufficiently hard to withstand the blows of the tappet. This also applies to all non-ferrous materials which have been suggested for valves, such as nickel-chromium and monel metal, the latter a natural alloy of nickel and copper containing at least 60 per cent of nickel and 23 per cent of copper, together with smaller amounts of other constituents. The wearing action on the tip of the stem is severe, and the tip must be hardened.

A great deal of experimental work on valve materials has been done in recent years, most of it with a view to the adaptability of the materials for aircraft engine valves, in which service the conditions are far more severe than in passenger car and truck service. The following comments on the properties of various alloys, which are based on published reports of such tests, must therefore be read with the fact in mind that in most lines of work the conditions are less exacting than those of the tests referred to.

Tungsten Steel—High tungsten steel (which is substantially the same as the so-called high speed tool steel) usually contains from 15 to 18 per cent of tungsten, 0.50-0.80 per cent of carbon, not over 0.30 per cent of manganese and from 3 to 4 per cent of chromium. The heat treatment for valves consists in heating to 2100° F., quenching in oil, reheating to 1325° F. and cooling in air, the tip being hardened by being reheated to a suitable temperature and allowed to cool in air while the head and stem are immersed in water.

When thus heat-treated, the steel has a tensile strength of 200,000 lbs. per sq. in. at normal temperature, which drops to 55,000 lbs. per sq. in. at 1200° F. and to 17,000 lbs. at 1600°. While very hard and brittle at normal temperature, the steel becomes quite ductile at high temperatures, and according to J. B. Johnson and S. A. Christianson of the U. S. Air Service, Engineering Division, it has an elongation of 60 per cent in one inch at 1600°. The Brinell hardness of the head is about 400.

As ccompared with other high-grade valve steels, tungsten steel scales appreciably when exposed to high temperatures. Scaling begins at about 1100°, and when the steel is exposed to a temperature of 1400° for 20 minutes, a heavy scale is formed. The most advantageous feature of tungsten steel is that it maintains its tensile strength remarkably well at high temperature, and for this reason it is preferred for racing engines to valves of a steel that has a lesser scaling tendency and which therefore would last longer than tungsten steel in a less arduous service.

Owing to the hardness of the steel at normal temperature, the tappet end of the stem will withstand the action of the tappet excellently, and the stem also resists wear in the guide well. As the steel is of the air-hardening type, the hardness of the head is not reduced materially by annealing in service.

Silicon-Chromium Steel—A steel of this type for valve purposes was introduced on the American market in 1922 by the Steel Products Co. of Cleveland, O., and has found wide use in heavy duty engines as well as in passenger car engines. Silicon-chromium steel in valves experimented with by the Air Service, Engineering Division, contained from 0.45 to 0.60 per cent of carbon, not over 0.50 per cent manganese, 2.75-3.75 per cent silicon and 7-9 per cent of chromium. The heat treatment for this steel consists in heating to 1925° F., quenching in oil, reheating to 1450° and cooling in air. The tip is hardened by heating to 1925° F., quenching in oil, reheating to 400° F. and quenching in oil.

This steel shows a tensile strength of as high as 208,000 lbs. per sq. in. at normal temperature, 42,000 at 1200° and 4000 at 1600°. It is more ductile than tungsten steel throughout the temperature range considered. The hardness of the head after the heat treatment outlined is 260 Brinell.

Silicon-chromium steels are particularly free from scaling, and in the test of the Air Service, the scaling was found to begin only at 1600°. The wear of the stem does not seem to be quite as satisfactory as with tungsten steel, the tip wears excellently. This is also an air-hardening steel and the head does not loose its hardness in service, at least not to any extent. One of the reasons for the excellent results obtained with steels of this class is said to be their comparative freedom from warping.

Cobalt-Chromium Steel—Among the valve steels tested by the Air Service was one of cobalt-chromium containing 1.12 per cent carbon, 1.03 per cent silicon, 12.17 per cent chromium, 0.81 per cent molybdenum and 2.47 per cent cobalt. This was found to give excellent results in service. The heat treatment consisted in heating to 1650° F., cooling slowly (50° per hour) to 1300° and then cooling to atmospheric temperature.

This steel has a tensile strength at atmospheric temperature of 124,000 lbs. per sq. in., which is reduced to 52,000 lbs. at 1200° and to 18,200 lbs. at 1600°. The Brinell hardness is 375, but is reduced by annealing in service to 275-300. The hardness of the tip after a 50-hour run in an aircraft engine was found to be higher than for any other valve steel. Scaling begins only at 1600°, and after the 50-hour run very little scale was found on the valves. The wear of the stems in cast iron guides is very satisfactory.

Owing to the different requirements in head and stem materials, heads of one metal are often welded to stems of another metal. Thus chrome-nickel steel heads have been welded to carbon steel stems by the electric butt-welding process. Chrome-nickel steel with high chromium and nickel content will serve for the head in the untreated state, having a fair degree of hardness and maintaining this well in service. The tensile strength of the steel at normal temperature is about 120,000 lbs. per sq. in. and drops to 60,000 lbs. at 1200° and 24,000 lbs. at 1600°. Scaling begins only at 1600°, and the amount of scale formed is in general very small.

This material is unsuited for stems, as not only will the tips not stand up under the wiping and hammering action, but the wear in the guide is quite rapid. For this reason carbon steel stems are welded to the chrome nickel steel heads, this having the further advantage that the carbon steel is lower in cost.

This process of building up valves by welding has been carried to its logical conclusion by Wills, who welds heads of a material resistant to scaling and pitting, to a stem of a material having good bearing qualities in cast iron under conditions of poor lubrication, and the latter to a tip of a material which is capable of withstanding the tappet action well.

French Tests of Valve Steels—Other tests with valves of different materials were reported in the *Revue de Metallurgie* (No. 22, 1925) by M. Mahoux. He points out the advantage of a low specific gravity in a valve material, as the shock imparted to the valve by the tappet at the beginning of the opening period, as well as the shock of seating, is directly proportional to the weight of the valve. This factor plays a rather important rôle in a comparison between tungsten and silicon-chromium steels, tungsten being a metal of high specific gravity.

The tests were made on an aircraft engine of 4.73-in. bore, 7.08-in. stroke, a compression ratio of 5.4 and running at 1860 r.p.m. The tests extended over 50 hours in 10-hour periods, the last 5 hours of the final run being under full throttle, a particularly severe test. At the conclusion of the bench test the valves were removed from the engine and inspected with respect to condition of the head, the stem and the guides.

In the accompanying table are given the compositions of the steels that proved most satisfactory in this severe test, and they are arranged in the order of merit.

Superior to all others was the behavior of Silcrome valve steel, the composition of which is given in the first column of the table. The critical point of this steel is approximately 1715° F., and in some samples even higher. A high critical point is decidedly favorable to durability in a valve material. Its specific gravity is 7.62, and therefore materially less than that of the high tungsten steel (No. 2), which has a specific gravity of 8.69, so that for the same dimensions the Silcrome valve of the test engine weighed about 1 ounce less than the tungsten steel valve. The four other steels listed in the table are tungsten steels.

Out of 91 American passenger-car engines manufactured in 1925, 29 had cast iron valve heads, 26 valve heads of silicon chromium steel, 15 of tungsten steel, 11 of nickel steel and 10 of carbon steel.

Order of Merit and Composition of Valve Steels Tested by Mahoux.

Material	1.	2.	3.	4.	5.
Carbon	0.44	0.72	0.51	0.7	0.42
Chromium .	9.04	4.96	2.08	5.0	2.47
Silicon	2.67	0.58	0.68	0.34	0.12
Phosphorus	0.012	0.009	0.018	....	Trace
Sulphur ...	Trace	0.018	0.052	....	.....
Nickel		Trace	.....	....	.....
Tungsten ..		17.15	13.0	17.0	15.87
Vanadium .	Trace	0.88	0.78	0.5	.....
Molybdenum	0.17	.....	.....	....	.....

Form and Dimensions of Head—The valve head should be made of the minimum thickness consistent with considerations of strength. In American practice the top surface of the valve is usually spherical and the bottom surface conical. This gives a head much thicker at the center (where the bending moment is a maximum) than at the edge. Poppet valves have been completely standardized by the Society of Automotive Engineers and the standardized dimensions are given in a table in the Appendix. The basic dimension is that of the bore of the port which is closed by the valve. Calling this d, the minimum diameter of the valve head is $d - 1/32$ in.; the width of the valve seat is

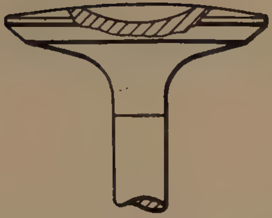


FIG. 181.—STANDARD STEEL VALVE.

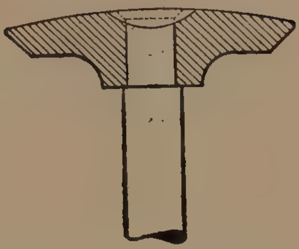


FIG. 182.—CAST IRON HEAD VALVE.

about 0.11 d; the maximum diameter of the valve head, 1.16 d and the valve stem diameter, $0.16\ d + 0.125$ in. In the S. A. E Standard the thickness of the head above the seat varies from 1/32 in. in the 1-in. size to 5/64 in. in the 3-in. size. The radius of the spherical top surface varies from 5 in. in the 1-in. size to 9-in. in the 3-in. size, and the radius at the junction of the head and stem from 1/4-in. in the 1-in. size to 9/16-in. in the 3-in. size. The angle made by the generatrix of the bottom cone of the head with a plane perpendicular to the stem is equal to 7° in valves below 2-in. nominal size and 12° in valves of 2-in. and over.

The seat on the valve head must project beyond the seat in the cylinder casting at both top and bottom, in order that it may not form a shoulder on the latter seat when it is ground in. A fairly wide seat is an advantage, as it helps to keep down the temperature of the head. The heat absorbed from the burning gases by the valve head has only two paths through which to flow off—down the valve stem to the valve guide and thence into the cylinder block and jacket, and through the valve seat directly into the block—and by far the greater portion passes off through the seat.

In Fig. 181 is shown the form of head generally used for steel valves in American practice, as fixed by the S. A. E. standard. Cast iron valve heads have the form shown in Fig. 182, if secured to the stems by a mechanical method,

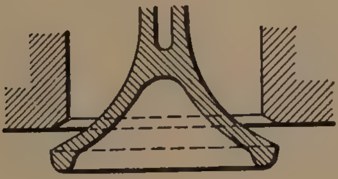


FIG. 183.—TULIP-TYPE VALVE

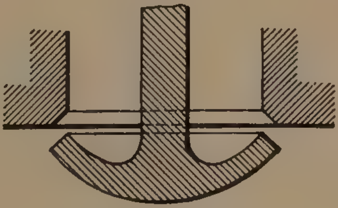


FIG. 184.—DOME-SHAPED VALVE.

whereas if they are butt-welded to the stems their form is very similar to that of heads of steel valves. The hub of the cast iron head shown in Fig. 182 has a tendency to restrict the valve passage, which is an objection to that form.

Unconventional Valve Heads—In order to provide a valve passage that favors stream-line flow and at the same time to ensure sufficient strength of the valve head at high temperatures, some designers, particularly in England, make use of what is known as the tulip type of valve, which is illustrated in Fig. 183. An improvement on this design, particularly adapted for use in engines having the valves in the head, is said to be the dome-shaped type illustrated in Fig. 184. This latter type of valve could not well be used in L-head engines, on account of the limited space in the valve pocket above the seat, but the claim that it affords greater strength with a certain amount of material seems to be well founded.

Degree of Filling—A gasoline engine cylinder in normal operation never draws as much charge per suction stroke as is represented by the piston displacement volume at atmospheric pressure and temperature. This latter amount is referred to as a complete charge. The ratio of the amount of air actually taken in per suction stroke to a complete charge is known as the volumetric efficiency. This factor decreases as the speed of the engine increases, increases and decreases with the size of the inlet valve opening area, and varies also with the valve timing and the cylinder wall temperature. Some tests made on a Franklin four-cylinder, 4 × 4 in. air-cooled engine at Cornell University, and reported in a paper read by Prof. R. C. Carpenter before the Society of Automobile Engineers at New York in January, 1911, showed that this engine, fitted with poppet valves of a clear diameter of 1⅜ in., opening 5° past dead center and closing 30° past dead center, had a volumetric efficiency when cold (that is, when driven by an electric motor) of 82 per cent at 1000 r.p.m. and 68 per cent at 1500 r.p.m. When run under its own power, with the inlet valves opening 5° past dead center and closing 30° past dead center, the volumetric efficiency was 68 per cent with applied phosphor bronze cooling fins and 62 per cent with cast iron integral cooling fins at 1000 r.p.m.; 63 per cent with applied phosphor bronze and 56 per cent with integral cast iron cooling fins at 1500 revolutions per minute. This clearly shows the effect of the cylinder wall temperature on the volumetric efficiency, the applied phosphor bronze fins evidently cooling more effectively than the integral cast iron fins.

Mean Gas Speed through Valves—In calculating the dimensions of valves, the calculations are often based upon a certain mean gas speed through the valve port, in feet per minute. It is obvious that (disregarding the presence of the valve stem) the mean gas speed through the valve port is to the mean piston speed as the piston head to the valve port area. But since the piston head area is proportional to the square of the bore and the valve port area to the square of the clear valve diameter, the mean gas velocity through the valve port is to the mean piston speed as the square of the bore is to the square of the clear valve diameter

$$v_v : v_p = b^2 : d^2,$$

From which it would follow that

$$v_v = v_p \, (b/d)^2$$

However, in view of the fact that the effective area of the valve port is reduced by the section of the valve stem, and more particularly because it is not the most restricted section of the inlet tract, it is preferable to base the mean gas speed on the area of the valve passage as calculated from equation (67). Calling this area a and the area of the piston head A, the theoretical mean speed of the gas through the valve is

$$v_v = v_p \, (A/a). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (68)$$

In modern engines v_v is generally equal to about 13,000 ft. p.m. at the speed of maximum output.

The volumetric efficiency is chiefly dependent upon two factors, the heating and consequent dilation of the charge upon entering the cylinder, and the wire-drawing at the inlet valve. In making calculations of volumetric efficiency, air at 60° F. and one atmosphere pressure is taken as the basis. Now, in order to vaporize the fuel it is necessary to heat the charge to 160°-180° F., and this necessarily expands it. Mixing the charge with the remanent gases in the combustion space and bringing it in contact with the combustion chamber walls may further increase its temperature, though, owing to the exceedingly short time available, it is hardly likely that any appreciable amount of heat is absorbed from the cylinder walls before the end of the inlet stroke.

If the temperature in the cylinder at the end of the inlet stroke were 180° F. then the volumetric efficiency, if the pressure within the cylinder at that moment were exactly one atmosphere, would be

$$\frac{(461+60)100}{461+180} = 81 \text{ per cent},$$

and this is about the volumetric efficiency which may be expected in an engine operating at normal working temperature but at a low speed.

With increase in speed the wire-drawing effect of the inlet valve (together with the resistance to gaseous flow of the rest of the inlet tract) prevents the attainment of atmospheric pressure in the combustion chamber at the end of the inlet period, and the volumetric efficiency is then lower. At the speed of maximum output the volumetric efficiency is generally between 65 and 70 per cent.

The horse power output of an engine depends upon two factors, the speed and the torque, and is directly proportional to the product of the two. If we assume that the efficiency of transformation of the heat energy into mechanical energy is constant, we can say the torque is directly proportional to the amount of combustible charge taken into the cylinder per cycle, or to the volumetric efficiency. Hence,

$$HP = cne,$$

where c is a constant; n, the speed in r.p.m. and e the volumetric efficiency. To find the condition under which the horse power becomes a maximum, we consider n and e as independent variables, differentiate the expression for horse power and make the first differential coefficient equal zero

$$dHP = c(nde + edn)$$

$$edn = -nde$$

$$\frac{dn}{n} = -\frac{de}{e}.$$

Therefore, the horse power is a maximum when the proportional rate of increase in speed is equal to the proportional rate of decrease in volumetric efficiency. In other words, when an increase of 1 per cent in speed is accompanied by a decrease of 1 per cent in the percentage of volumetric efficiency.

At the time when high volatility fuels were still in common use, that required practically no heat supply to the inlet manifold, the volumetric efficiency at low engine speeds with the engine at normal working temperature was between 86 and 88 per cent, but with our present hot-spot systems we cannot expect more than about 82 per cent. With no pre-heating of the charge the volumetric efficiency will be about 70 per cent for a theoretical gas speed of 11,000 ft. p.m. through the valve port, corresponding to about 13,000 ft. p.m. through the valve opening itself. With the gas entering the engine at say 180° F. this would be reduced to about 65 per cent. The figures here given must be regarded as averages, as there are other factors besides the temperature of the entering charge and the theoretical speed of the gases through the valve port that affect this efficiency.

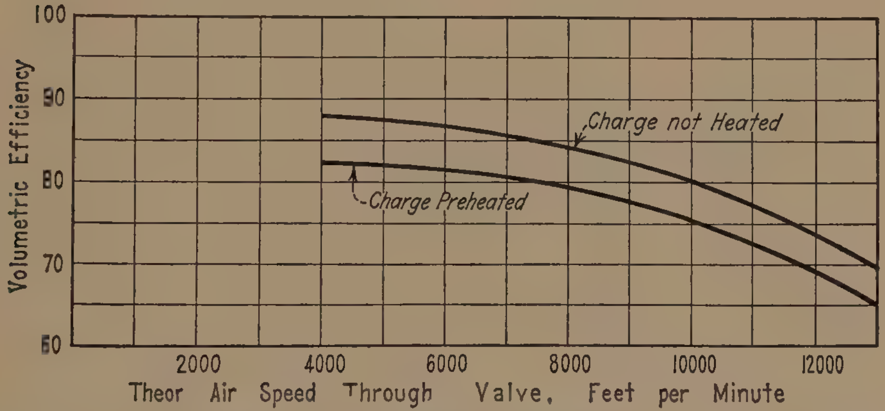


FIG. 185.—VARIATION OF VOLUMETRIC EFFICIENCY WITH GAS SPEED THROUGH VALVE.

Valve Guides—Sometimes the guides for the valve stems are cast integral with the cylinders, but more frequently they are inserted. Casting them integral is the least expensive construction, but it is open to the objection that, owing to unavoidable inaccuracies in molding, the holes through them are often considerably out of center, and, besides, an unbushed guide cannot be refitted when worn. A fairly accurate fit of the stem in the guide is necessary particularly in the case of the inlet valve, as air will be drawn through the guide during the inlet stroke if there is any play; if the amount of play varies at the different guides, the mixture received by the different cylinders will be unequally diluted, and it is impossible to correct this by carburetor adjustment. The exhaust valve guides cannot be properly lubricated, and wear in these, therefore, is comparatively rapid.

Separate guides are made of cast iron or bronze, and they are inserted either from the top (that is, through the valve seat) or from the bottom. The latter method appears to be more common. The guide is provided with a flange which abuts against a boss on the cylinder casting and serves as a spring support. It is a good plan to make this bearing quite narrow and have the flange project beyond it on the outside, as in that case less heat will be transmitted to the valve spring and there is less danger of its temper being drawn.

If the guide is inserted from above it is sometimes slightly tapered and inserted in a tapered hole; otherwise it is formed with a slight shoulder. The two methods of inserting the guides are illustrated in Fig. 186. The wall of the guide is made from ⅛ to 3/16 in. thick, and its length is made from 1.75 to 3 times the clear diameter of the valve, the shortest guides being used with overhead valves.

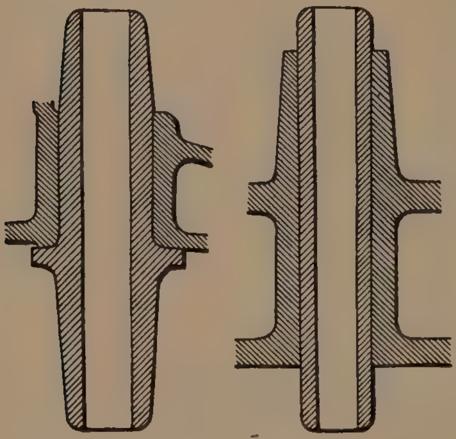


Fig. 186.—Valve Stem Guides.

Valve Timing—The valves are operated by cams on a shaft which turns at one-half the speed of the crankshaft, so that each valve is opened and closed once during two revolutions of the crankshaft. For operation at very low speed the inlet valve should begin to open and should close substantially at the beginning and end of the inlet stroke, respectively, and the exhaust valve should open a trifle before the end of the power stroke and close at the end of the exhaust stroke. However, in order that the engine may operate satisfactorily at high speed, it is necessary to alter the valve periods considerably. The exhaust valve must open earlier and close later, and the inlet valve must open later (in order not to overlap the exhaust opening) and close later. The lag of inlet closing and the lead of exhaust opening should be greater in proportion as the engine is intended to run at higher rotary speed. For a truck engine intended to run at not more than 1400 r.p.m. and having large sized valves, the following valve timing may be used:

Exhaust opens 45° ahead of bottom dead center
Exhaust closes 10° past top dead center
Inlet opens 15° past top dead center
Inlet closes 30° past bottom dead center

For high-speed passenger-car engines running up to 3000 r.p.m. the following timing may be used:

Exhaust opens 50° ahead of bottom dead center
Exhaust closes 6° past top dead center
Inlet opens 10° past top dead center
Inlet closes 45° past bottom dead center

It should be pointed out that if an engine is timed to give the very best output at high speeds it will not run so satisfactorily at low speed, and will not be as flexible. This is due to the fact that when the exhaust valve opens very early, some of the power otherwise available at low speed is lost through the exhaust, and when the inlet valve closes very late, some of the charge drawn in during the suction stroke will be forced out again during the beginning of the compression stroke. The cylinder, therefore, will not receive as large a charge as it might, and will not utilize to the fullest advantage the charge it does receive.

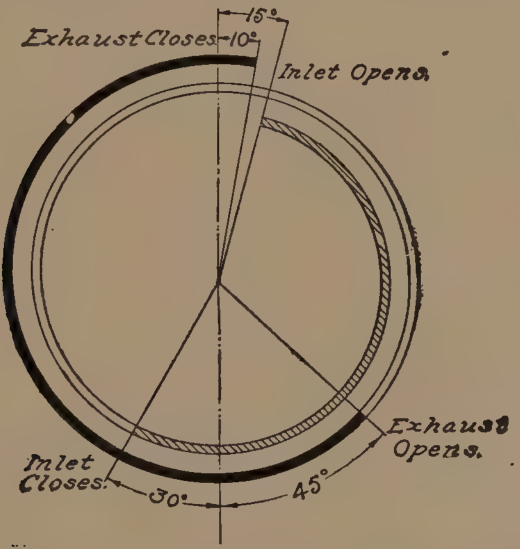


Fig. 187.—Valve Timing Diagram for Moderate-Speed Engines.

Some tests made at the Laboratory of the Automobile Club of France on a four-cylinder 100 by 140 mm. Renault engine with variable inlet timing, indicated that the output is a maximum and the specific fuel consumption a minimum when the inlet closes with a lag of 60°. However, so great a lag is rarely used in practice. It is worth pointing out in this connection that the most desirable lags and leads depend to a certain extent upon the rate of valve opening and closing.

Cams and Cam Followers—The rate at which the valve is opened and closed depends upon the cam outline and upon the type and size of cam follower employed. From the standpoint of gas flow it is, of course, desirable that the valve should open and close very quickly, and remain fully open for the greatest possible length of time. However, the valve gear must operate quietly, and in order to do this it must lift and drop the valves more or less gradually.

There are three common types of cam follower. The most extensively used in this country is the mushroom type; next comes the cam roller and finally the rounded or V type.

There are also three types of cam, viz., the tangential cam, the convex flanked cam, used together with a mushroom type follower (called for short the mushroom type cam), and the constant acceleration cam, whose flank is concave. Either roller or round nose followers are used with the first and third types of cam. The tangential cam has been in use longest, and gives very satisfactory results, the only objection to it being that it requires a rather stiff spring. The mushroom type cam does not call for so stiff a spring, and opens the valve very quickly, thus giving a better volumetric efficiency, but it strikes the cam follower rather a sharp blow when first coming in contact with it, especially when the adjustment is not very close. The constant acceleration cam in one sense is a compromise between the two other types, raising the valve less abruptly than the mushroom cam, and decreasing its speed more gradually, in consequence of which it does not require as strong a spring as the tangential cam. With this cam the valve is first accelerated at a uniform rate until it attains its maximum speed; from that point on the valve is uniformly decelerated, until at the point of maximum lift it has lost all speed. Disregarding the fact that there is a slight change in the pressure of the valve spring between the points of half lift and full lift, this cam possesses the advantage that at high speed the spring force is fully utilized throughout the period of deceleration.

However, the denomination "constant acceleration" does not by any means completely fix the form of the cam, even for a given arc of opening and maximum lift, for the periods of acceleration and deceleration are independently variable and the two together may be equal to or less than one-half the period of valve opening.

Laying Out Tangential Cam—In Figs. 198–190 are shown the three forms of cam, and the methods by which they are laid out may be briefly described as follows: Suppose that the cam is to lift the inlet valve, and that the latter is to begin to open at 10° past the top dead center, and to close 40° past the bottom dead center. This makes the total period of opening equal to

$$180° + 40° - 10° = 210°$$

of crank motion, or 105° of cam motion. The cam follower, of course, begins to lift at the moment when the radius at

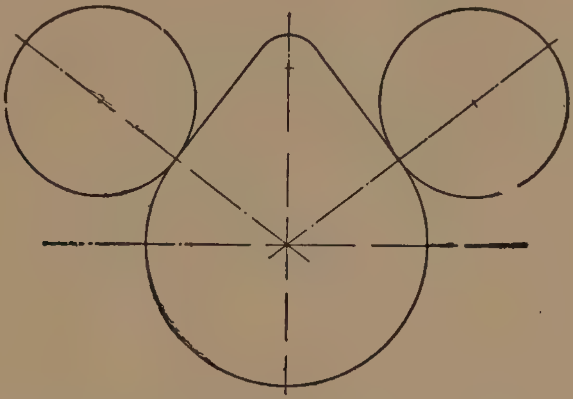


Fig. 188.—Tangential Form of Cam.

the point of tangency between the cam incline and the base circle is in line with the axis of the follower. Therefore, we first draw the base circle, which must be slightly larger in diameter than the camshaft and the exact diameter of which is determined by a method of calculation to be explained further on. Then we draw two radial lines, making with each other an angle of 105°, and a third radial line bisecting this angle. Along the latter, from the circumference of the base circle outward, we lay off a distance corresponding to the maximum lift of the valve. In the case of the tangential cam, at the points of intersection of the other two radial lines with the circumference of the base circle, we draw tangents to the base circle, and then with the point of the compass on the central radial line we draw a small circular arc such that it passes through the point of maximum lift and that the tangents to the base circle are also tangent to it. This completes the cam outline. We have then a cam which will raise the valve during the first half of the period of cam opening and lower it during the second half, there being no period of dwell in the position of full opening.

In case the period of valve opening is considerably over 100° of cam motion, as is often the case with exhaust cams, provision is made for a period of dwell in the position of full lift. We then draw a circular arc through the point of full lift, determined as explained in the foregoing, concentric with the base circle, and at the intersections of this arc with the tangents to the base circle draw rounding arcs to a radius the dimensions of which will be discussed further on.

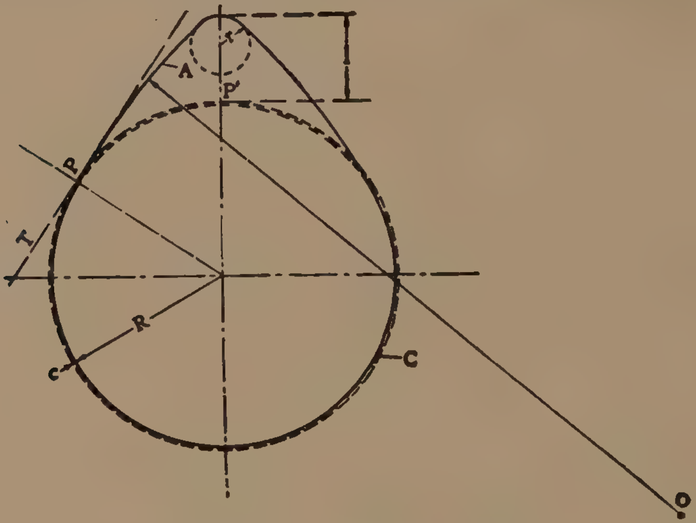


Fig. 189.—Mushroom-Type Cam.

Laying Out Mushroom Type Cam—The mushroom type of cam follower is slightly simpler in construction than the roller cam follower, and it also reduces the number of wearing parts by eliminating the roller pin. But since it substitutes sliding for rolling motion at the cam surface, it is doubtful whether it reduces wear in the valve gear as a whole.

The method of laying out a cam for a mushroom type follower is as follows (Fig. 189): First draw the base circle of radius R and the top circle of radius r, the two radii being assumed. The two centers must be placed at a distance

$$D = R + l - r,$$

l being the valve lift. Now lay off a tangent *T* to the base circle at a point *P* such that its angular distance from point *P′* (where the line connecting the centers of the top and base circles cuts the base circle) is equal to one-half the valve lift period angle *a*. Then draw an arc *A* which is tangent to the top circle and the tangent *T*. The center of this arc is found to be located at *O*, which point, naturally, lies on an extension of the base circle diameter passing through the point of tangency *P*. This arc forms the flank of the cam nose. The radii of the base circle, top circle and flank curve, and the lift and opening angle are mutually related, and these relations will be discussed further on. With mushroom cams there is no period of dwell in the full open position for either inlet or exhaust cams, and both kinds of cam therefore are laid out the same way.

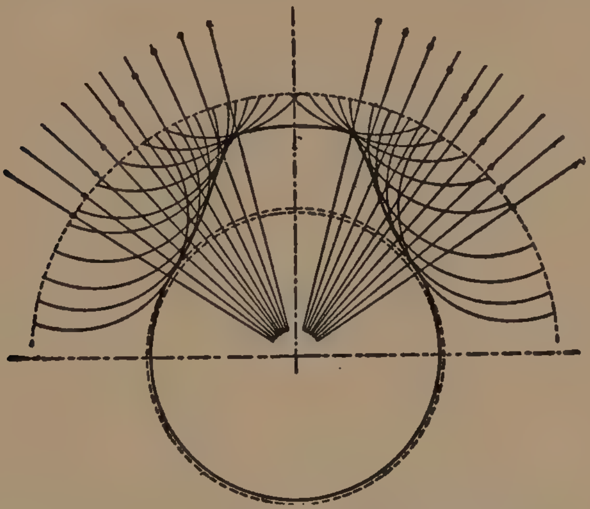


Fig. 190.—Constant Acceleration Cam.

Laying Out Constant Acceleration Cam—Constant acceleration cams are laid out somewhat differently from the other two types, as the inclines of these cams are not plain geometrical lines, but curves of varying radius. After having laid off the base circle and the three radial lines denoting the beginning, middle and end of the opening period, respectively, we divide the opening period into a certain number of equal parts, and draw radial lines through all division points. Then, by means of a formula to be developed later, we calculate the valve lift for each of these positions. Next we lay off the position of the cam roller center for each of these positions along the corresponding radial line, remembering that the distance of the center of the cam roller from the center of the base circle is equal to the sum of the radius of the base circle, the radius of the roller circle and the lift. We then draw circles of the same radius as the cam roller around the various cam roller centers thus found and then draw a curve which is tangent to all of these circles and to the base circle, which constitutes the cam outline. In the case of exhaust valve cams it is advisable to have a period of dwell in the position of maximum opening, but the application of the above method remains the same.

The kinematics of the tangential and mushroom types of cam were investigated by George S. Bowen (*Internal Combustion Engineering,* London, Oct. 15, 1913), Otto M. Burkhardt (*The Horseless Age,* Jan. 28 and Feb. 4, 1914) and A. L. Nelson (*The Horseless Age,* Sept. 23-Oct. 21, 1914), and the following treatment is based on their methods:

Valve Speed and Acceleration with Tangential Cam—In Fig. 191 is shown the outline of a tangential cam with the roller in the position where the lift begins. While in reality in a vertical engine the center of the cam roller is always vertically above the center of the cam base circle, in all of the diagrams of cam mechanisms in this chapter the whole mechanism is swung around through such an angle as to bring the radius of the cam base circle to the point of tangency of the cam incline into a horizontal position, this making the geometrical relations in most cases clearer. For the purpose of this discussion we will disregard the small clearance ("the thickness of a sheet of ordinary writing paper"), which the cam roller must have when the valve is seated and assume that the valve begins to lift as soon as the tangent strikes the cam roller. The lifting period evidently divides into two parts, which must be dealt with separately, viz., (1) when the cam roller is in contact with the straight portion of the cam incline, and (2) when it is in contact with the circular portion of the top of the cam—the top circle or rounding circle. Assuming uniform rotation of the cam, we must develop equations for the acceleration or deceleration of the cam follower and valve, then determine the maximum value of the deceleration, and having found this we must design a spring capable of imparting to a weight equal to that of the reciprocating parts of the valve mechanism (including one-half the weight of the spring), an acceleration equal to the maximum deceleration determined by the cam outline for the maximum speed of the engine. The use of such a spring insures that the cam follower always remains in contact with the cam outline, thus obviating noisy operation.

Referring to Fig. 191 it will be seen that

$$AB \cos \theta = R + R_1.$$

Hence

$$AB = \frac{R + R_1}{\cos \theta}.$$

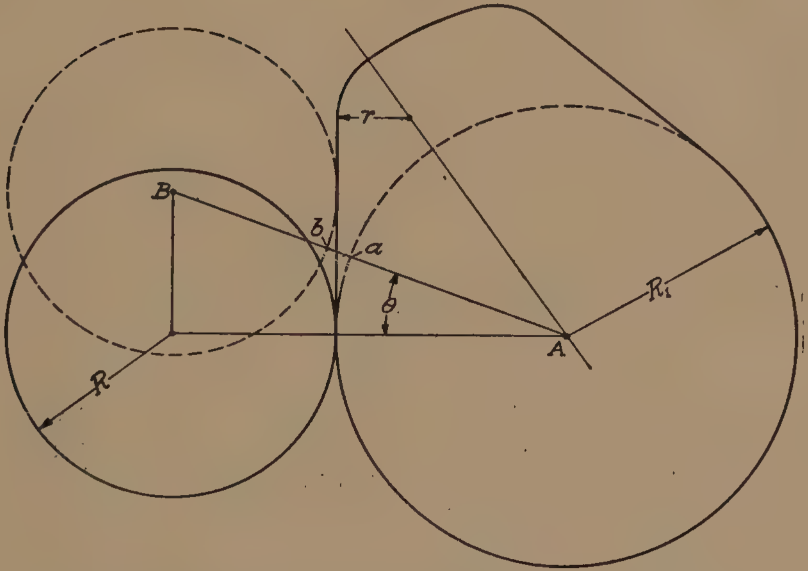


FIG. 191.—TANGENTIAL CAM, ROLLER ON STRAIGHT FLANK.

The dotted circle shows the roller on the straight part of the cam, and *ab* evidently represents the lift at this point, after the cam has turned through an angle θ.

$$l = ab = AB - (R + R_1)$$

$$= \frac{R + R_1}{\cos \theta} - (R + R_1)$$

$$= (R + R_1)\left(\frac{1}{\cos \theta} - 1\right).$$

To get the speed or rate of lift we differentiate this expression with respect to θ, which gives

$$dl = (R + R_1)\frac{\tan \theta}{\cos \theta}d\theta.$$

The actual speed of lift, however, is represented by

$$\frac{dl}{dt}.$$

hence,

$$v = (R + R_1)\frac{\tan\theta d\theta}{\cos\theta dt}.$$

But

$$\frac{d\theta}{dt} = \frac{\theta}{t} = \frac{2\pi N}{60}.$$

Hence

$$v = (R + R_1)\left(\frac{2\theta N}{60}\right)\frac{\tan\theta}{\cos\theta}. \quad . \quad . \quad . \quad . \quad (69)$$

To find the acceleration we differentiate this latter expression, which gives

$$a = (R + R_1)\left(\frac{2\pi N}{60}\right)^2\left(\frac{1 + \sin^2\theta}{\cos^3\theta}\right). \quad . \quad . \quad (70)$$

It will be seen from this that as long as the cam roller remains on the straight part of the cam incline the valve is accelerated, because the sine increases with the angle and the cosine decreases until for an angle of 90°, which is, of course, far beyond the practical limit, the cosine becomes zero, and hence the theoretical acceleration infinite. This further shows that deceleration of the valve is confined to that period of cam rotation during which the cam roller is in contact with the circular portion of the cam incline of radius r.

Referring now to Fig. 192, which shows the cam roller on the circular portion of the cam incline, since the sines of the angle of an oblique triangle are proportional to the opposite sides

$$\frac{AC}{\sin(180° - \beta)} = \frac{R + r}{\sin\gamma}.$$

But

$$\gamma + \delta + (180° - \beta) = 180°,$$

hence

$$\beta = \gamma + \delta.$$

But

$$\sin(180° - \beta) = \sin\beta,$$

and hence

$$= \sin(\gamma + \delta).$$

Therefore

$$\frac{AC}{\sin(\gamma+\delta)} = \frac{R+r}{\sin\gamma},$$

from which it follows that

$$AC = (R+r)\frac{\sin(\gamma+\delta)}{\sin\gamma}$$

$$= (R+r)\left(\frac{\sin\gamma\cos\delta + \cos\gamma\sin\delta}{\sin\gamma}\right)$$

$$= (R+r)(\cos\delta + \cot\gamma\sin\delta).$$

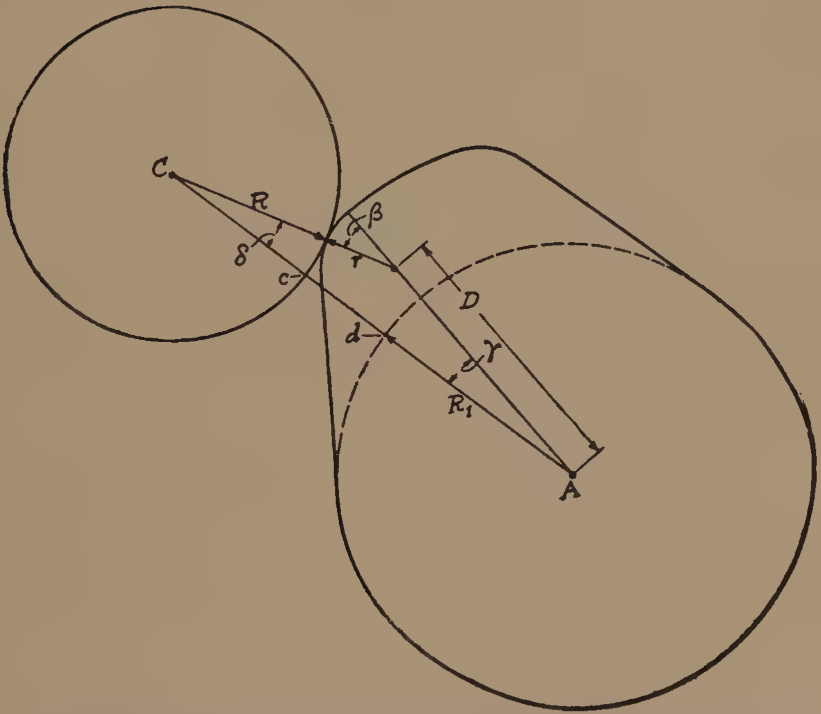


Fig. 192.—Tangential Cam, Roller on Rounding Circle.

Also, by the same proposition quoted above,

$$\frac{D}{\sin\delta} = \frac{R+r}{\sin\gamma},$$

so that

$$\sin\delta = \frac{D\sin\gamma}{R+r},$$

and

$$\cos \delta = \sqrt{1 - \frac{D^2 \sin^2 \gamma}{(R+r)^2}}.$$

Substituting these values for sin δ and cos δ in the above expression for AC, we get

$$\begin{aligned} AC &= (R+r)\left[\sqrt{1 - \frac{D^2 \sin^2 \gamma}{(R+r)^2}} + \cot \gamma \frac{D \sin \gamma}{R+r}\right] \\ &= \sqrt{(R+r)^2 - D^2 \sin^2 \gamma} + D \cot \gamma \sin \gamma \\ &= \sqrt{(R+r)^2 - D^2 \sin^2 \gamma} + D \cos \gamma. \end{aligned}$$

The lift of the valve at this point, when the cam must turn through an angle γ till the valve is fully lifted, is

$$\begin{aligned} l = cd &= AC - (R + R_1) \\ &= \sqrt{(R+r)^2 - D^2 \sin^2 \gamma} + D \cos \gamma - (R + R_1). \end{aligned}$$

Differentiating this expression with respect to the time, we get

$$v = \frac{dl}{dt} = \left[\sqrt{\frac{D^2 \sin \gamma \cos \gamma}{(R+r)^2 - D^2 \sin^2 \gamma}} + D \sin \gamma\right]\frac{2\pi N}{60}. \qquad (71)$$

Differentiating again to find the acceleration we obtain

$$a = \frac{dv}{dt} = -\left[D^2 \frac{D^2 \sin^4 \gamma - (R+r)^2 (2 \sin^2 \gamma - 1)}{\{(R+r)^2 - D^2 \sin^2 \gamma\}^{3/2}} + D \cos \gamma\right]\left(\frac{2\pi N}{60}\right)^2. \quad . \quad . \quad . \qquad (72)$$

This is evidently a maximum when $\gamma = o$, that is, at the moment the valve attains its full lift. The expression then reduces to the comparatively simple form

$$a = -\left(\frac{D^2}{(R+r)} + D\right)\left(\frac{2\pi N}{60}\right)^2 \text{in. p. s. p. s.} \quad . \quad . \quad . \qquad (73)$$

By now dividing by 12 to reduce to feet per second per second and inserting the above value of a in the equation

$$F = \frac{Wa}{g}$$

we get for the spring force required to hold the valve down to the cam

$$F = \frac{W}{12g}\left(\frac{D^2}{R+r} + D\right)\left(\frac{2\pi N}{60}\right)^2. \quad . \quad . \quad . \qquad (74)$$

Relation Between Parts of Tangential Cams—It is not hard to see that there must be a definite relation between the radius of the base circle, the radius of the rounding circle and the angle of complete lift of a tangential cam. In an inlet cam there usually is no period of dwell at full lift, the valve beginning to drop again as soon as it has been fully lifted. In that case the angle of complete lift is equal to $\phi/2$ or one-half the angle of opening. We then have (Fig. 193)

$$D \cos \frac{\phi}{2} = R_1 - r$$

$$D = l + R_1 - r$$

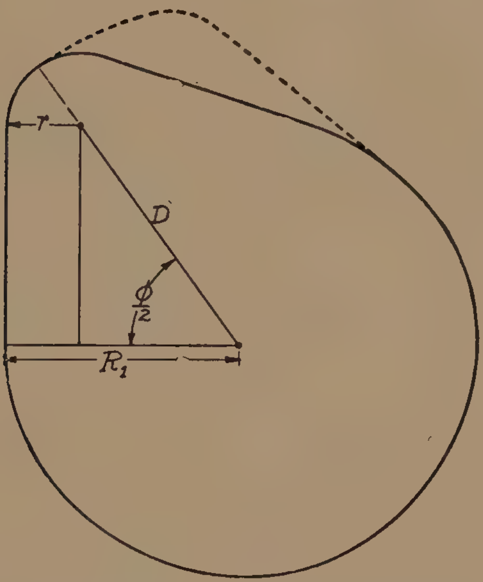


FIG. 193.—TANGENTIAL CAM, RELATIONSHIP OF PARTS.

Substituting for D in the first of the above equations we get

$$R_1\left(1 - \cos \frac{\phi}{2}\right) = r\left(1 - \cos \frac{\phi}{2}\right) + l \cos \frac{\phi}{2},$$

$$R_1 = r + l \frac{\cos \frac{\phi}{2}}{1 - \cos \frac{\phi}{2}}.$$

This same equation can, of course, also be applied to exhaust cams in which there may be a period of dwell in the full open position (as shown by dotted curve), but in that case the angle of complete lift must be substituted for one-half the angle of opening.

In plotting the curve of lift by the tangential cam it is desirable to know the angle θ_m corresponding to the angular motion of the cam from the point of beginning of lift to the point where the cam follower first comes in contact with the top circle. The cam is shown in this position in Fig. 194, and it will be seen that

$$\tan \theta_m = \frac{D \sin \frac{\phi}{2}}{R + R_1}. \quad . \quad . \quad . \quad . \quad . \quad (75)$$

Of course, it is not absolutely necessary to know the value of this angle in order to plot the curve of lift, because by starting to plot from the points of beginning of lift and maximum lift, according to the formulae for the lift on the straight and circular portions of the cam incline, respectively, we reach a point where the two portions of the lift curve intersect and this, of course, is the point corresponding to the angle θ_m, and completes the lift curve.

We may now figure out an example. Assume that for an inlet cam the period of opening is to be 100° so that

$$\frac{\phi}{2} = 50°.$$

Also, that the lift is to be 5/16 inch, and the radius of the rounding circle 3/16 inch. In that case

$$R_1 = 3/16 + 5/16\left(\frac{\cos 50°}{1 - \cos 50°}\right) = 3/4 \text{ inch.}$$

From this we find the value D to be

$$D = 3/4 + 5/16 - 3/16 = 7/8 \text{ inch.}$$

If we assume the valve reciprocating parts to weigh one pound and the cam roller to have a radius of $\frac{1}{2}$ inch, then the pressure of the valve spring in the position of full opening for a crankshaft speed of 2400 r.p.m. must be

$$\frac{1}{12 \times 32.16}\left(\frac{7/8 \times 7/8}{11/16} + 7/8\right)\left(\frac{6.28 \times 1200}{60}\right)^2 = 81 \text{ lbs.}$$

Valve Speed and Acceleration with Mushroom Cam—So far as that portion of the "mushroom" type cam incline is concerned which depends on the radius of the top circle, the same analysis applies as in the case of the tangential cam. The cam follower having a flat surface, it is equivalent in its action to a roller of infinite radius. By making $R = \infty$ in the equation for the spring pressure required, the term

$$\frac{D^2}{R + r}$$

vanishes, and the equation becomes

$$F = \frac{WD}{12g}\left(\frac{2\pi N}{60}\right)^2$$

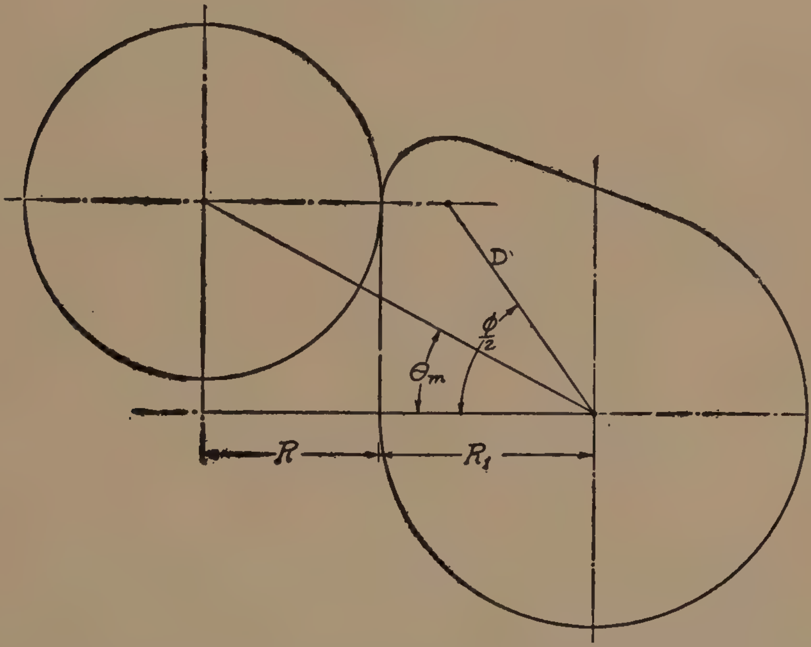


FIG. 194.—TANGENTIAL CAM, ROLLER AT JUNCTION OF STRAIGHT AND CURVED PORTIONS.

It is, however, desirable to also investigate the acceleration while the follower is on the flank curve. Referring to Fig. 195 the lift is represented by

$$ab = bc - ac$$

$$bc = df = R(1 - \cos \theta)$$

$$ac = \qquad R_1(1 - \cos \theta)$$

subtracting

$$l = (R - R_1)(1 - \cos \theta)$$

Differentiating with respect to t

$$v = (R - R_1) \sin \theta \frac{d\theta}{dt}$$

$$= (R - R_1) \sin \theta \left(\frac{2\pi N}{60}\right)$$

Differentiating again with respect to time we get

$$a = (R - R_1) \cos \theta \left(\frac{2\pi N}{60}\right)^2 . \quad . \quad . \quad . \quad (76)$$

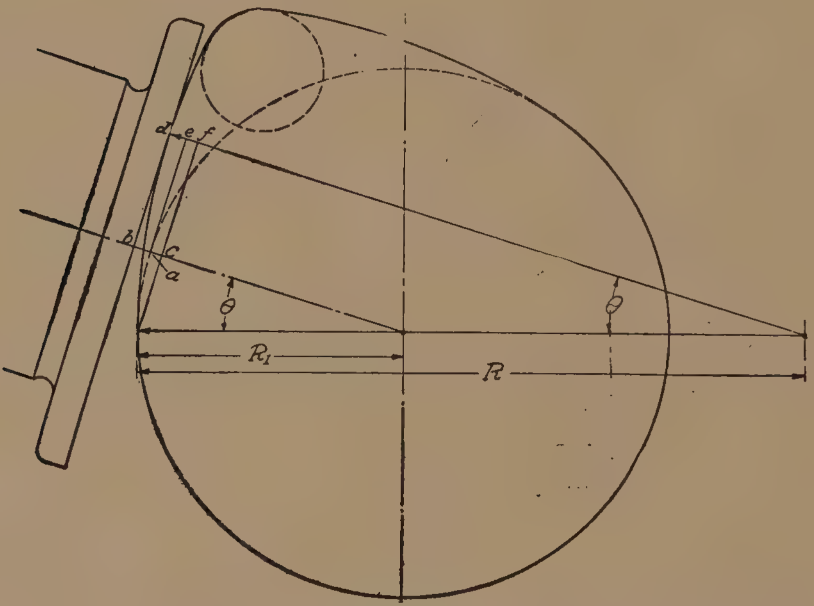


FIG. 195.—MUSHROOM-TYPE CAM, FOLLOWER ON FLANK CURVE.

This gives positive results for values of θ up to 90°, far beyond the practical limits, and shows that deceleration takes place only during that portion of the cam period while the follower makes contact with the top circle. The circular motion of the cam from the beginning of valve lift to the point where the follower contacts with the cam at the junction of the two circular arcs making up its incline may be found from the equation (see Fig. 196)

$$(R - r) \sin \theta_m = D \sin \frac{\phi}{2},$$

$$\sin \theta_m = \frac{D}{R - r} \sin \frac{\phi}{2} . \quad . \quad . \quad . \quad . \quad . \quad (77)$$

In order to be able to draw the full curve of lift for the mushroom cam we must derive also an expression for the lift when the cam follower contacts with the top circle. By referring to Fig. 197 it will be seen that the lift for that part of the rotation of the cam while the follower contacts with the top circle may be expressed by the equation

$$l = D\cos\gamma + r - R_1$$

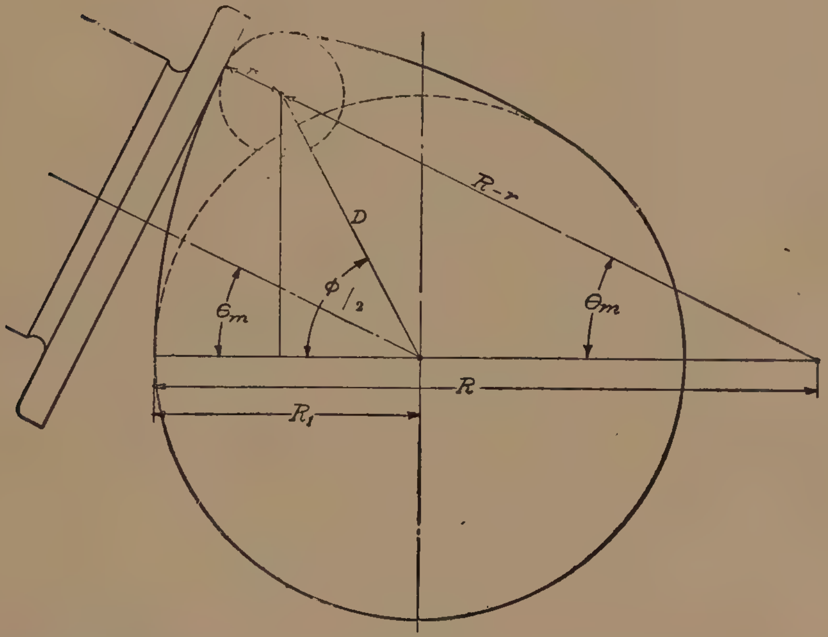


FIG. 196.—FOLLOWER AT JUNCTION OF FLANK AND TOP CIRCLES.

When the valve is fully lifted the value of γ is *nil*, and the maximum value of this angle is equal to

$$\gamma_m = \frac{\phi}{2} - \theta_m. \qquad (78)$$

Determination of Radius of Flank Circle—The radius of the flank circle of a cam for a mushroom type follower depends upon the radii of the base and top circles and the angle of cam motion corresponding to the full lift. Generally with "mushroom type" cams there is no dwell of the valve in the full open position, the valve beginning to close as soon as it has attained its maximum lift. In other words, the angle of cam motion corresponding to the full lift of the valve is equal to half the angle of valve opening $(\phi/2)$. Referring now to Fig. 196

$$(R - r)^2 = \left(D \sin \frac{\phi}{2}\right)^2 + \left(R - R_1 + D \cos \frac{\phi}{2}\right)^2$$

$$R^2 + r^2 - 2Rr = D^2 \sin^2 \frac{\phi}{2} + R^2 + R_1{}^2 - 2RR_1$$

$$+ 2RD \cos \frac{\phi}{2} - 2R_1D \cos \frac{\phi}{2} + D^2 \cos^2 \frac{\phi}{2}$$

$$= D^2 + R^2 + R_1{}^2 - 2RR_1 + 2RD \cos \frac{\phi}{2} - 2R_1D \cos \frac{\phi}{2}$$

$$2RR_1 - 2Rr - 2RD \cos \frac{\phi}{2} = D^2 - r^2 + R_1{}^2 - 2R_1D \cos \frac{\phi}{2}$$

$$R = \frac{D^2 - r^2 + R_1{}^2 - 2R_1D \cos \frac{\phi}{2}}{2\left(R_1 - r - D \cos \frac{\phi}{2}\right)} \quad . \; . \; . \; . \quad (79)$$

This value becomes infinite and the cam therefore becomes a tangential cam (which can be successfully used only with a roller or round nosed follower) if

$$R_1 = r + D \cos \frac{\phi}{2}.$$

Spring Force Required with Constant Acceleration Cam—Let l be the lift and a the angle of cam motion during which the valve is completely lifted. The time required for turning this angle is

$$\frac{a}{\frac{2\pi N}{60}} = \frac{60a}{2\pi N} \text{ seconds.}$$

Since the acceleration is constant the maximum speed is twice the average, viz.,

$$\frac{2l}{\frac{60a}{2\pi N}} = \frac{4\pi N l}{60a} \text{ inches per second,}$$

or

$$\frac{\pi N l}{180a} \text{ feet per second.}$$

Now let us suppose that deceleration takes place in a fraction f of the total time of lift or

$$\frac{60fa}{2\pi N} = \frac{30fa}{\pi N}.$$

Then the rate of deceleration is evidently

$$a = \frac{\dfrac{\pi Nl}{180a}}{\dfrac{30fa}{\pi N}} = \frac{\pi^2 N^2 l}{5400 fa^2}. \quad . \quad . \quad . \quad . \quad . \quad (80)$$

Inserting this value of a in the equation for accelerating force we get

$$F = \frac{n^2 N^2 Wl}{5400 fga^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (81)$$

In this equation a must be inserted in terms of radians.

As the lifting angle is generally expressed in degrees, by letting a represent degrees and incorporating π^2 and g in the constant we get

$$F = \frac{N^2 Wl}{5.36 fa^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (82)$$

We may now summarize the formulæ for the spring pressures required and will at the same time reduce them to their simplest and handiest form by incorporating all known factors in the numerical constants.

For tangential cams

$$F = \frac{WN^2}{35200}\left(\frac{D^2}{R+r} + D\right) \text{ lbs.}$$

For mushroom follower cams

$$F = \frac{WN^2 D}{35200} \text{ lbs.}$$

For constant deceleration cams

$$F = \frac{WN^2 l}{5.36 fa^2} \text{ lbs.}$$

where

W = weight of valve reciprocating parts (including one-half of spring) in pounds.

D = distance between center of base circle and center of top circle or rounding circle in inches.

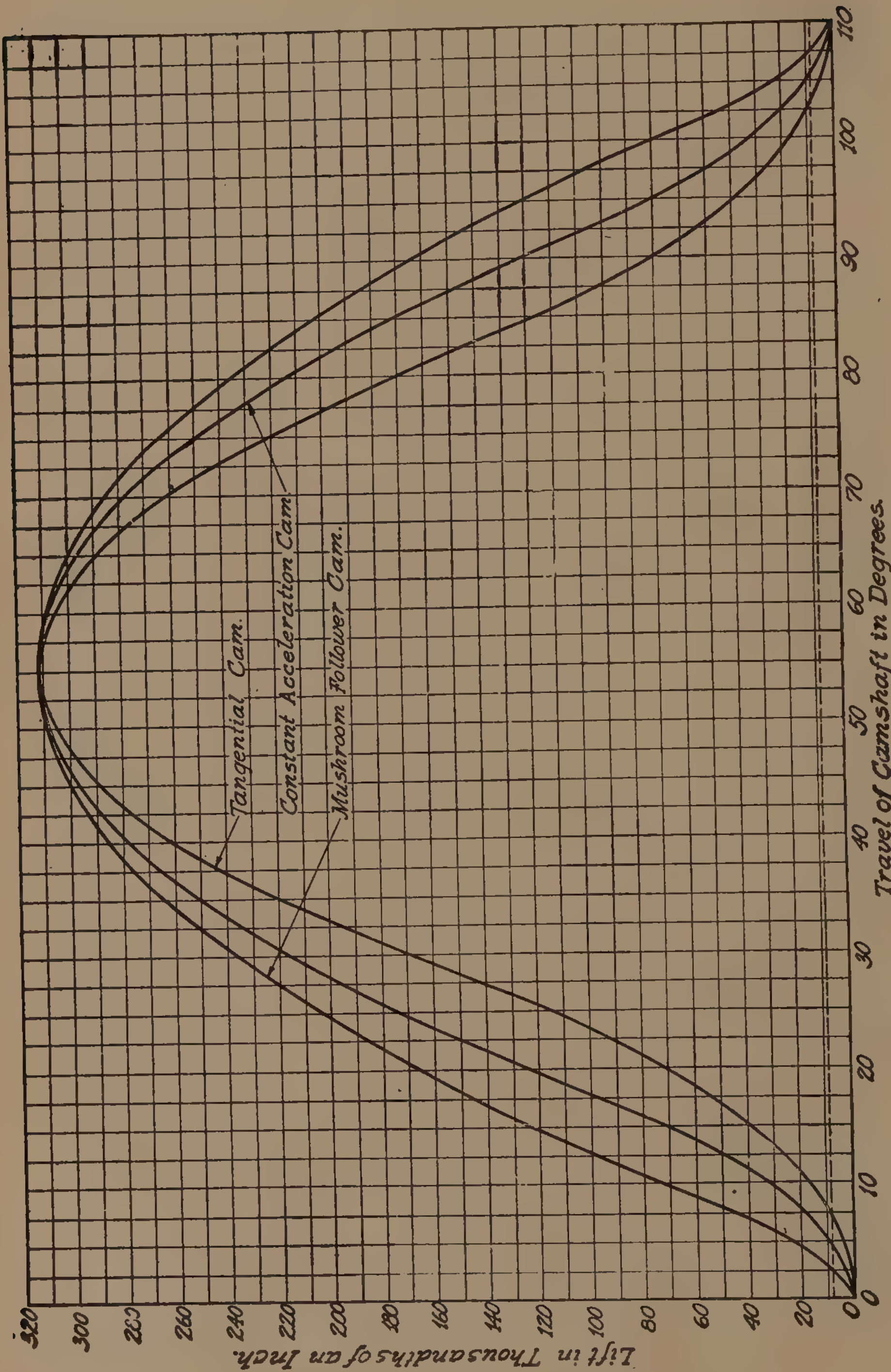


FIG. 198.—CURVES OF LIFT FOR THE THREE FORMS OF CAM.

R = radius of cam roller in inches.
r = radius of top circle or rounding circle in inches.
N = speed of camshaft in r.p.m.
l = lift of valve in inches.
a = angle of camshaft motion in degrees required to completely lift the valve.
f = fraction of this angle during which the valve decelerates (f is made greater than 0.5 to reduce the spring force required).

In the case of the tangential and mushroom-type follower cams the spring pressure obtained by the formulae is that which the spring must exert when the valve is fully lifted. When the valve is closed the spring pressure, of course, will be less as the spring is then less compressed. In the case of the constant deceleration cam the spring must exert a pressure equal to that calculated by means of the formula when the valve has attained its maximum speed and is beginning to decelerate. At that moment the valve has been lifted a distance

$$l = (1 - f) L \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (83)$$

Comparison of Valve Lift Curves—In Fig. 198 are shown valve lift curves for the three types of cam—tangential, mushroom and constant acceleration—which permit of certain comparisons being made. It should be pointed out, however, that no absolute comparison is possible, as with each type of cam there are certain assumed values which change the form of the curve; these variables are entirely different in character in the different cam types and it is impossible to assign to them values that might be called strictly equivalent. In the figure the lift of the valve and the period of opening are the same in every case. These dimensions are such as might be used for the inlet cam of a high-speed engine. In every case there is no dwell at full lift. The data for the three cams are summarized in Table IX, page 330.

It will be seen from the figure that with the mushroom follower cam the valve accelerates very quickly, and after 15° of cam motion the lift is more than three times as great as with the tangential cam, which gives the smallest acceleration to the valve. The constant acceleration cam, as here chosen, produces a lifting curve lying midway between the other two. This latter curve may be widely varied by varying the factor f. For instance, if f were made equal to 0.5, so that the

Table IX.—Data of Valves of Three Types

Tangential	Mushroom	Const. Accel.
$l = 5/16$ inch	$l = 5/16$ inch	$l = 5/16$ inch
$r = 3/16$ inch	$r = 1/8$ inch	$\frac{\phi}{2} = 55°$
$R = 1/2$ inch	$R_1 = 5/8$ inch	f † = 2/3
R_1 * = 0.607 inch	R * = 6.66 inches	
$\frac{\phi}{2} = 55°$	$\frac{\phi}{2} = 55°$	

* Calculated from other data.

† Proportion of total period of lift that valve decelerates.

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acceleration during the first half of the lift were equal to the deceleration during the last half, then the lift curve for the constant acceleration cam would correspond rather closely to that of the tangential cam. The steepness at the beginning of the lift curve is a measure of the blow dealt the cam follower by the cam when first coming in contact with it.

For any given cam motion from the point where the valve begins to lift, the lift is greatest for the mushroom cam, hence the volumetric efficiency would be expected to be greatest with that cam. Moreover, from the standpoint of strength of spring required, the mushroom-type cam also ranks first in merit. When the camshaft has still 20° to travel to the full lift position, the valve has yet to pass through a vertical distance of about 0.048 inch in the case of the mushroom-type cam, 0.062 inch in the case of the constant acceleration type cam, and 0.092 inch in the case of the tangential cam. The average speed of the valve for the remainder of the lift is therefore greatest in the case of the tangential, and least in the case of the mushroom-type cam. Since the speed of the valve in all cases reduces to zero during these 20° of cam motion, it may be inferred that the deceleration is greatest in the case of the tangential cam. Of course, the best way of comparing the cams in this respect is by calculating the spring pressures required with each cam, by means of the formulæ derived in the foregoing.

If we assume that the weight of the valve reciprocating parts in each case is 1 lb., that the radius R of the cam roller is 0.5 inch, and that the maximum speed of the camshaft is 1,200 r.p.m., we get for the valve-spring pressure required in the different cases:

Tangential cam

$$\frac{1 \times 1200 \times 1200}{35200}\left(\frac{0.732 \times 0.732}{0.6875} + 0.732\right) = 62 \text{ lbs.}$$

Mushroom-type cam

$$F = \frac{1 \times 1200 \times 1200 \times 0.8125}{35200} = 33.3 \text{ lbs.}$$

Constant acceleration cam

$$F = \frac{1 \times 1200 \times 1200 \times 0.3125}{5.36 \times 0.666 \times 55 \times 55} = 41.6 \text{ lbs.}$$

Speed and Acceleration Curves—While the lift curves give some idea of the characteristics of the different cams, these latter are brought out much more plainly by speed and acceleration curves, as shown for the same cams in Figs. 199 and 200, respectively. These curves were drawn from data calculated by the equations for speed and acceleration developed in the foregoing for the different types of cam. Referring to Fig. 199, the speed of the valve with the constant acceleration cam increases and then decreases at a uniform rate. Thus the speed curve or speed line for this cam forms with the base line a triangle, and as distances along the base line measure time as well as angular motion, the area of this triangle represents the total lift. The total lift is the same for all three cams, hence the area enclosed by the speed curves and the base line is the same in all three cases. Since the maximum speed attained by the mushroom type cam follower is less than that attained by the follower of the constant acceleration cam, it follows that at least a part of the speed curve of the former must be concave toward the base line, and for a similar reason the speed curve of the tangential cam must be convex toward the base line. With the mushroom type of cam the maximum speed of lift is attained in less than 6° of cam rotation, which points to a very intense force active between cam and follower during the first part of the lift. With the constant acceleration cam the maximum speed is attained after a cam motion of 18° 20′ and with the tangential cam after a cam motion of a little over 28°. With all cams there is a sudden change from a rapid increase in speed to a rapid decrease.

The differences in the actions of these various types of cam can be further studied in Fig. 200. With the constant acceleration cam, for instance, the acceleration proceeds at a constant rate during the first one-third of the period of lift, and is followed by deceleration at one-half that rate during the last two-thirds of the lift period. It is the maximum value of the deceleration that determines the strength of valve spring required.

Valve Clearance—Owing to the fact that when an engine heats up in service the exhaust valve stem expands more than the cylinder barrel, it is necessary to allow some clearance between the elements of the valve lifting mechanism when the engine is cold and the valve is seated, as otherwise the valve would fail to seat when the engine heated up and would be quickly destroyed. In the foregoing theoretical discussion of the cam mechanism we disregarded this clearance; that is, we assumed that when the valve was seated its stem was in contact with the cam follower and the latter was in contact with the cylindrical portion of the cam or the base circle.

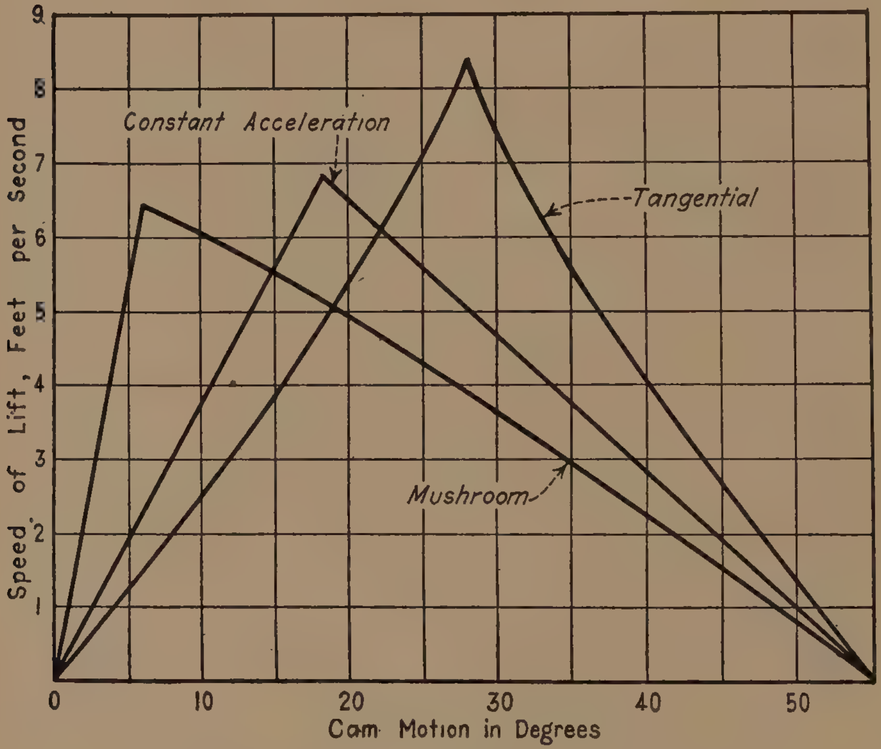


Fig. 199.—Curves of Speed of Valve Lift.

We can provide clearance in two different ways. One consists in shortening the follower (by means of the adjusting screw usually provided), so that if it remained in contact with the valve stem it would not come down to the base circle. The effect of such an adjustment on the operation of the cam may be explained by reference to Fig. 198, where the horizontal dotted line near the base line represents a clearance of 0.008 in., the amount usually allowed for the exhaust valve. When there is clearance, a shock is produced when this is taken up and the various elements of the mechanism come in direct contact. The magnitude of this shock is proportional to the angles made by the different lift curves with the clearance line. It will be seen from the figure that the mushroom-type cam crosses this line at by far the greatest angle, consequently this cam would be expected to be the most noisy, followed by the constant acceleration cam as here proportioned and by the tangential cam, in that order.

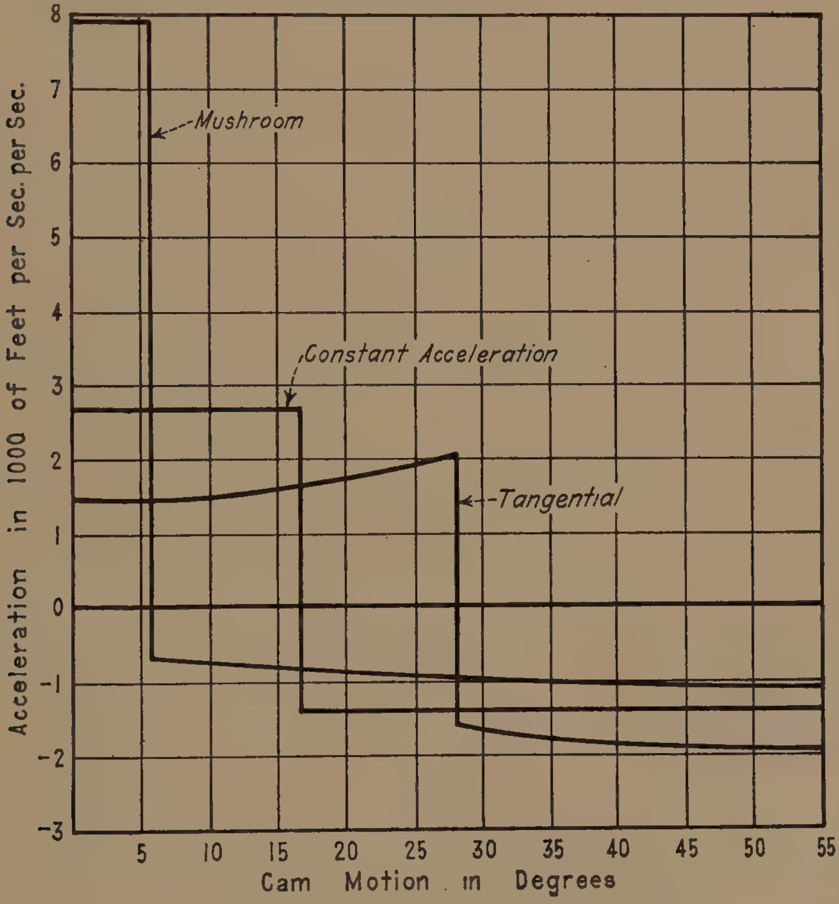


FIG. 200.—VALVE ACCELERATION DIAGRAM.

If the clearance were nearly taken up, which might be expected to be the case if the engine were operated at full load for some time, the cam action would evidently be more silent. But silent operation is desired particularly when the engine is running slowly under small load, as there is then little noise from the exhaust and from the transmission mechanism, and any noise from the cams therefore is the more noticeable.

For this reason the other method of providing clearance above referred to is generally employed. This consists in making the cylindrical portion of the cam of a radius less than the radius of the theoretical base circle and providing at the beginning and end of the cam surface proper a ramp or quieting curve. This is essentially a spiral surface, or, when developed, an inclined plane, so that, no matter at what portion of this surface the follower contacts when the clearance is taken up, the angle or rate of lift is always the same, and hence also the quietness of operation. For instance, since about 0.008 in. clearance is required for the exhaust valve, the ramp might extend over an arc of 8° on the cam surface and the radius increase uniformly at the

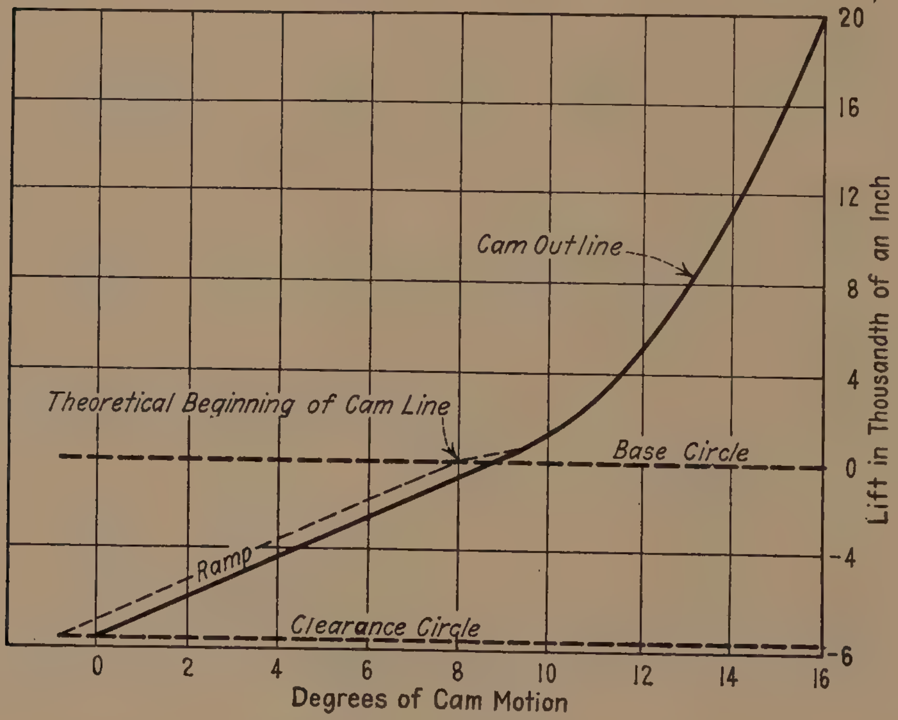


FIG. 201.—DEVELOPMENT OF CAM RAMP OR QUIETING CURVE.

rate of 0.001 inch per degree. The ramp may either join the cam outline at its point of contact with the base circle, or it may join it a little higher up, where the inclination of the cam outline is the same as that of the ramp, so that the ramp forms a tangent to the cam profile. A portion of a cam outline and a ramp, developed on a straight line, are shown in Fig. 201. This is not a true development, vertical distances being enlarged much more than horizontal (or circumferential) distances, so as to bring out the principles involved more clearly. Both methods of construction are indicated. That shown in dotted lines is more commonly used in this country.

Side Thrust on Push Rod Guide—Among the factors determining the value of any valve construction is the average side thrust of the cam follower on its guide, which should be as low as possible. The vertical reaction between the cam surface and the cam follower is equal to the sum of the spring pressure and the inertia force. The latter is positive during the first half of the lift and the last half of the drop, and negative during the last half of the lift and the first half of the drop. However, the reaction is vertical only at no lift and at full lift. For all intermediate positions the reaction is at an angle, as indicated in Fig. 202. Calling the vertical pressure P, it is evident that the pressure or reaction at the contact surface between cam and cam follower is

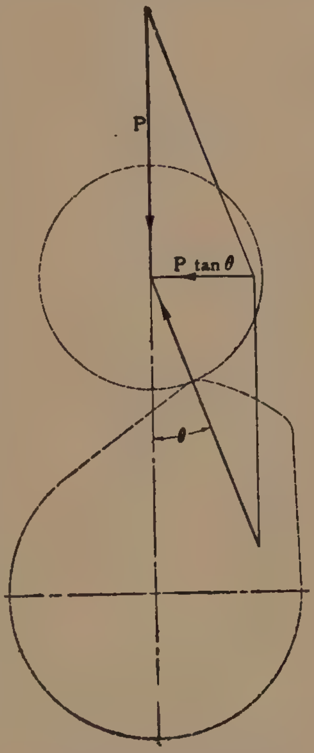


Fig. 202.—Cam Side-Thrust Diagram.

$$P_r = \frac{P}{\cos\theta}$$

and the side thrust on the push rod guide

$$P_s = P\tan\theta.$$

Calculation of Springs—The maximum safe pressure of coiled wire springs and the deflection under a given load are obtained by means of the following formulæ:

$$W = \pi \frac{Sd^3}{8D}. \qquad (84)$$

$$F = \frac{8nPD^3}{Ed^4}. \qquad (85)$$

Where

D = mean diameter of the coil,
W = maximum safe load in pounds,
F = compression of spring,
d = diameter of wire,
n = number of coils in spring,
S = maximum safe fiber stress of material.
E = torsional modulus of elasticity,
P = load in pounds.

The torsional modulus of elasticity may be taken at 12,000,000 and the maximum safe working stress may be assumed to be 50,000 pounds per square inch. In general the outside diameter of the spring is made equal to the clear diameter of the valve, and the length of the spring under its initial pressure twice its outside diameter. These proportions can be used for guidance in laying down values for some of the factors in the above equations.

To show the application of the above formulæ we will calculate a spring for a valve with a lift of ⅜ inch. We will assume that the mean diameter (from center to center of wire) of the coil is 1½ inches and that a spring pressure of 80 pounds is required when the valve is fully lifted. We will calculate this spring so that it exerts 60 pounds pressure when the valve is closed. Since the lift of the valve is ⅜ inch, 20 pounds pressure compresses the spring ⅜ inch.

The necessary diameter of wire we find from equation (84). Inserting the assumed values in this equation we have

$$80 = \frac{3.14 \times 50{,}000 \times d^3}{8 \times 1.5}$$

$$d = \sqrt[3]{\frac{960}{157{,}000}} = 0.183 \text{ inch.}$$

This is approximately the diameter of a No. 7 (Birmingham gauge) steel wire, which we will adopt. The actual diameter of this wire is 0.18 inch. Now, to find the number of turns required, we take into consideration the fact that the total deflection is to be ⅜ inch for 20 pounds load.

Inserting in equation (85)—

$$\frac{3}{8} = \frac{8 \times 20 \times 1.5^3 \times n}{12{,}000{,}000 \times 0.18^4},$$

$$n = \frac{\frac{3}{8} \times 12{,}000{,}000 \times 0.18^4}{8 \times 20 \times 1.5^3} = 9 \text{ approximately.}$$

It was stated above that the length of the spring under its initial pressure is usually about twice its outside diameter. We will assume that the drawing in our case shows that the spring can be 3 inches long. Then, since the spring when the valve is closed is under 60 pounds pressure and it compresses 3/8 inch for 20 pounds pressure, it follows that its initial compression is

$$\frac{60 \times \frac{3}{8}}{20} = 1\frac{1}{8} \text{ inches.}$$

The spring should therefore be wound to a length of

$$3 + 1\frac{1}{8} = 4\frac{1}{8} \text{ inches.}$$

If the results obtained under the first assumptions are not satisfactory, different assumptions can be made and the calculation made over again. With relatively short springs, as used on overhead valve engines, the difference in the spring pressures with the valve open and closed, respectively, will be greater than here assumed.

Spring Rests—At A in Fig. 203 is shown a spring rest which with slight variations is used by many American makers. The conical collar which surrounds the lower end of the stem and fits into a rounded groove on same, is split in halves, so it can be put around the stem, and it is firmly pressed against the latter by the conical center of the pressed steel spring washer. The chief advantage of this construction is undoubtedly that the collar is pressed against the stem by the spring and no looseness can develop at this point. Sometimes the end of the stem is provided with three narrow and shallow grooves instead of a single larger one.

The construction shown at B is also in common use. The valve stem is provided with a transverse hole or slot in which is inserted a key. The key forms a support for the washer and is held from endwise motion by a downwardly turned flange on same. An upward flange on the washer centers the spring. In design C the valve stem is reduced in diameter near its lower end and a steel collar with a radial slot is slipped over this reduced portion to support the washer, which is of the shape shown, and is stamped from heavy sheet metal. At D is shown an arrangement which has been used to a considerable extent in France. It is identical with B except for the fact that the sheet metal washer has no flange for retaining the key, which latter is somewhat wider and provided with a slot at the middle of its under side, whereby it is retained in position. Valve stem tips have been standardized by the S. A. E. (see Appendix).

Design of Push Rods—If the valves are located in side pockets of the cylinders, they are operated from the cams through the intermediary of push rods. The latter usually consist of a cylindrical steel part moving in a cast iron guide secured to the crankcase. The push rod carries the cam roller or other type of cam follower at its lower end, and is provided with adjusting means at its upper end. Two typical push rods are shown in Fig. 204.

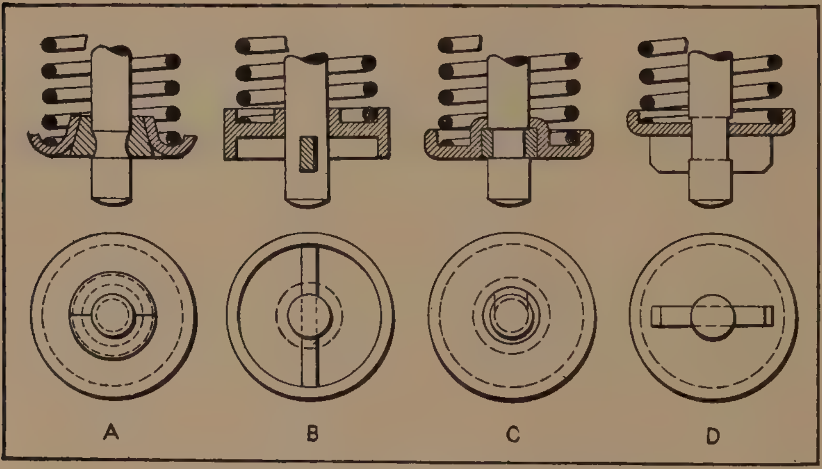


Fig. 203.—Valve-Spring Supports.

In the design of these members, lightness is an important consideration, since the strength of the valve spring required, and the shock and noise produced, will be directly proportional to the weight of the parts moved by the cam, which include the push rod. In Fig. 204, A is the body portion of the pushrod, which is drilled out from the top for the sake of lightness. The lower end is slotted transversely to receive the roller B, which is carried on the roller pin C. The roller extends into slots in the guide, whereby it is prevented from rotating in the latter. At its upper end the push rod is internally threaded to receive the adjusting screw

E. To the hexagonal head of this screw a wrench can be applied, and after adjustment is made the screw is locked by the check nut *F*. Screw *E* is also drilled out, with the object of reducing the weight. The head of the screw should be hardened, so it will not be marred or upset by the continual hammering it receives as it strikes the end of the valve stem.

Pushrod guides may be secured to the crankcase by means of two studs passing through lugs cast integral with them, and suitable nuts, or the guides of adjacent valves may be held down by means of X-shaped yokes through which passes a stud screwed into the crankcase wall midway between the guides. A more modern method is to arrange the push rod guides in clusters, each cluster usually containing four guides in the case of a four-cylinder engine and either four or six in the case of a six-cylinder. These clusters are bolted against a finished surface on the crankcase or the cylinder-crankcase block. The advantage of the cluster arrangement is that it effects economies on the work of machining. The construction shown in Fig. 205 is typical.

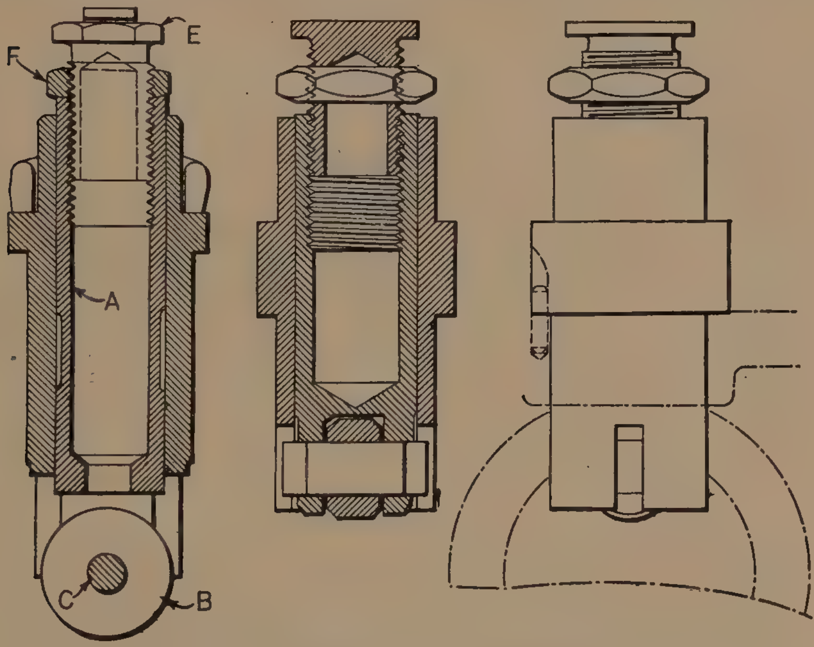


FIG. 204.—TWO DESIGNS OF ROLLER-TYPE PUSH ROD.

Clearance Required—A certain clearance in the valve-operating mechanism is required because in the operation of the engine the valve stems, especially those of the exhaust valves, are raised to a higher temperature than the cylinders, whose temperature is limited by the water jacket. Consequently, if the push rods were adjusted while the engine was cold so that there was practically no clearance in the valve-operating mechanism, the valves would not seat after the engine had heated up. In L-head engines the clearance allowed between push rod and valve stem usually amounts to 0.004 inch in the case of the inlet valve, and 0.008 inch in the case of the exhaust valve.

Proportions of Push Rod Elements—The push rod bearing surface must be brought down as near to the cam as practicable, because the load on it is an overhanging load, and if the overhang were great the pressure near the lower end of the bearing would be very intense and there would be rapid wear. The dimensions of the various elements of the push rod are proportional to the valve port diameter. Calling this *d*, the push rod diameter is made 0.5*d;* the length of the push rod bearing, 2*d;* the roller diameter, 0.63*d;* the roller width, 0.25*d* and the roller pin diameter, 0.25*d*. In the case of mushroom-type cam followers, the diameter of the mushroom head is made equal to the diameter of the cam base circle or a little smaller. With mushroom type cam followers it is a good plan to offset the follower lengthwise from the middle of the cam, so the follower will rotate in its bearing and no groove will wear in its foot. Such a follower is shown in Fig. 206.

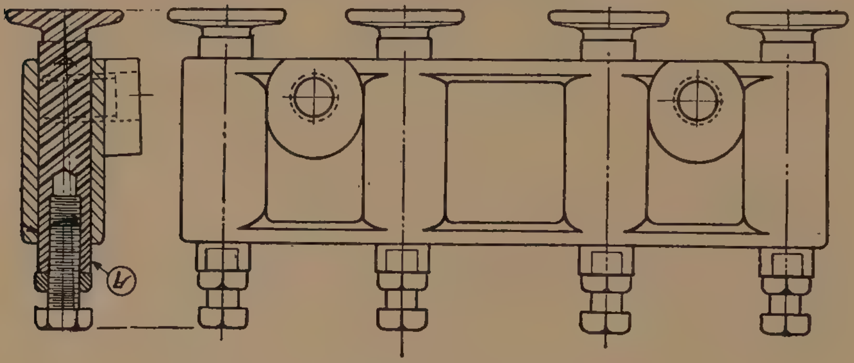
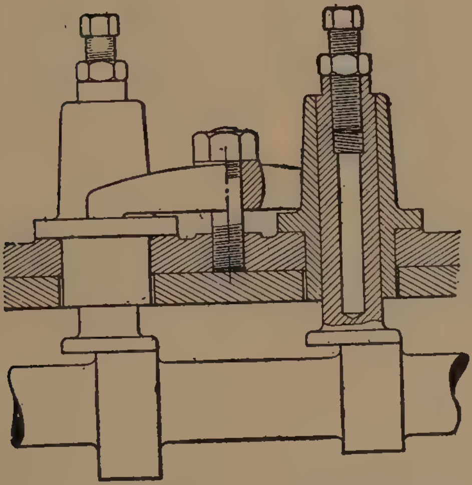


FIG. 205.—PUSHROD GUIDE CLUSTER.

Mushroom-type cam followers are usually made of low carbon steel, which is carbonized and then hardened. Recently it has come to be a common practice to weld a foot of cast iron to a stem of soft steel, chilling the cast iron and thus hardening it.

Cam Levers—In order to reduce the side thrust on the push rod, one-armed levers are sometimes interposed between the cam and the push rod, as illustrated in Fig. 207. These levers are so arranged that they occupy a position perpendicular to the push rod axis when the valve is half lifted. The horizontal component of the cam thrust then comes on the lever and is taken up by the lever pivot. However, the push rod is not entirely relieved of side thrust, because the free end of the lever moves in an arc of a circle, instead of straight up and down, and the bottom end of the push rod has to slide on it. If that part of the lever on which the push rod bears is given such a curvature that the push rod is always at right angles to the contact surface, then the side thrust is equal to the product of the upward thrust on the push rod by the friction coefficient This is considerably less than the side thrust of the cam, but since the object of the cam lever is only partly accomplished and since it adds to the complication, this arrangement has lost considerable of its one time popularity. The aim now seems to be to provide the pushrod guide with a liberal surface, bringing it down as near to the cam as possible and provide for copious lubrication. Bell crank levers are sometimes used between cam and push rod in V



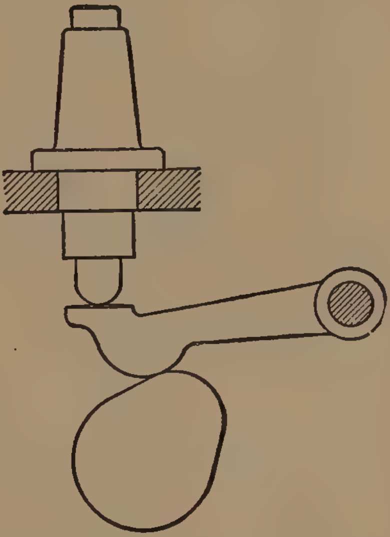


Fig. 206 (left).—Mushroom-Type Cam Follower and Guide Secured by Yoke.

Fig. 207 (right).—Rocker Lever between Cam and Pushrod.

engines to permit of the operation of similar valves of oppositely located cylinders by means of a single cam.

Overhead Valves—The earliest automobile engines with valves in the head had integral cylinder heads, which required the valves to be set in cages, as already explained. This arrangement did not prove entirely satisfactory, owing to inadequate cooling of the valve heads. When the removable cylinder head was introduced these cages were no longer necessary, and the valve-in-head engine has gained materially in favor since that time. Valves in the head are operated either by tappet rods extending up the side of the cylinders, or by means of an overhead camshaft. When tappet rods are used, these extend up through an enclosed space, and the rocker arms, etc., at the top of the engine are enclosed by a valve cover, usually in aluminum. When the whole valve mechanism is thus enclosed, not only are any noises produced by it muffled, but the joints can all be effectively lubricated and all bearings are protected against dust. In multi-cylinder overhead valve engines the rocker arms are usually mounted on a hollow shaft, to which is connected a lead from the pressure lubricating system. At the center of each rocker arm bearing an oil hole is drilled through the wall of the hollow shaft, so that oil will feed to the bearing surface, and sometimes a hole is drilled

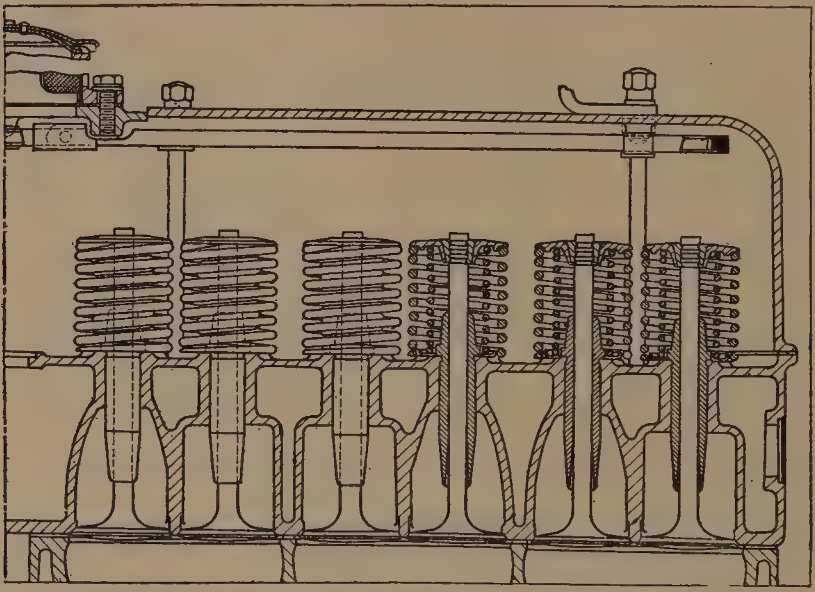


FIG. 208.—VALVES IN CYLINDER-HEAD, WITH DOUBLE SPRINGS.

lengthwise of the rocker arm from the rocker bearing surface to the point of contact with the valve stem. By placing the breather pipe on the valve cover and establishing communication with the crankcase by means of the tappet rod passages, an oil-misty atmosphere is created in the valve chamber, and the lubrication of all bearings is provided for.

With tappet rods extending up the side of the engine, as shown in Fig. 209, the weight of the moving parts naturally is greater than with ordinary valve gears for valves located in side pockets, and every effort should therefore be made to eliminate all unnecessary weight. The tappet rods are preferably made of steel tubing of an outside diameter of about 3/8 in. Into the lower end of the tube can be fitted a thrust pin with a half-round head, which head has a bearing in a suitable socket screwed into the hollow push rod. At the top another pin with hardened head is let into the tube, against which bears the spherical point of the clearance-adjusting screw. Owing to the fact that the contact surfaces of the joints are very liberal, and of such shape that they naturally retain any oil getting onto them, there is little wear. The rocker levers on top of the engine are rigidly supported by a hollow shaft carried in a number of substantial brackets.

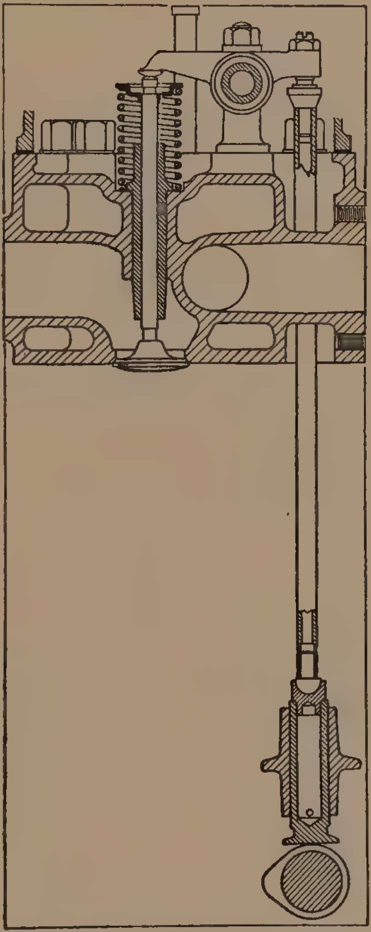


Fig. 209.—Overhead Valve Operated through Side Rod.

Overhead Camshafts — The simplest and most direct method of actuating valves in the cylinder head is by means of an overhead camshaft. It reduces the necessary weight of valve reciprocating parts, thus making it possible to get along with lighter springs and to increase the maximum speed. Unfortunately, it has been found difficult to obtain with this construction the degree of silence in operation which the American purchaser of quality cars expects.

Practically all aircraft engines have overhead camshafts, and in Europe a moderate number of both passenger car and commercial vehicle engines are provided with them. In this country three manufacturers of passenger cars use overhead camshaft engines—Wills, Stutz and Duesenberg. It appears that one reason for the apparently less quiet operation of the engine is that the source of the noise (the cam gear) is directly underneath the bonnet and the noise is therefore more readily transmitted to the passengers' ears than when it occurs down in the crankcase. It has been found that a contributing factor to noise in the valve gear is the discontinuity of the torque; that is, when the nose of a cam has passed a cam follower, the pressure of the valve spring causes the cam and its shaft to snap forward, thereby taking up the clearance between gear teeth. To prevent this, it is customary in overhead camshaft engines to burden the camshaft with additional load, such as the fan, water pump or generator. In the Wills-St. Claire engine the overhead camshaft is provided with a brake to provide a constant drag and prevent the shaft from snapping ahead. Referring to Fig. 210, at the rear end the camshaft carries a steel cone keyed to it, and on the extended hub of this slides a cast iron cone, the two cones being pressed into frictional engagement with a double female cone of bronze keyed in the camshaft housing.

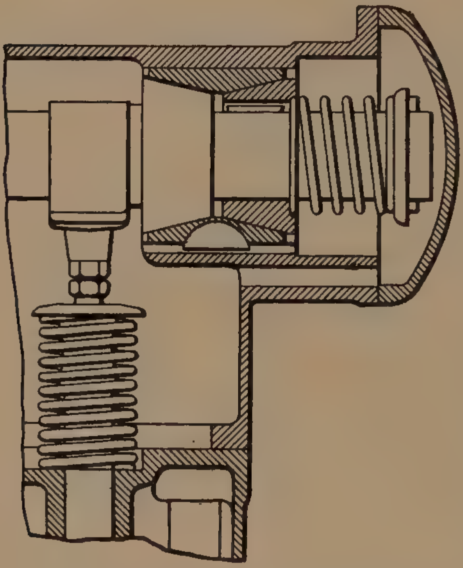


Fig. 210.—Brake on Overhead Camshaft (Wills).

Direct Actuation—Valves can be operated from overhead camshafts either directly or through the intermediary of rocker arms or bell cranks. Both systems are used on aircraft engines, the direct method of actuation having been used particularly on the Hispano-Suiza. The direct actuated valve of the Stutz engine is shown in Fig. 211. Naturally, considerable side thrust on the valve stem is produced by the cam action, and means must be provided for taking this up without causing undue wear. The stems of the valves have their upper ends threaded to receive the valve operating nut and piston, which latter unit is guided by a cast iron sleeve that is piloted in and bolted to the upper face of the cylinder head. Particular pains are taken to get the cast iron guide sleeve concentric with the bore of the valve stem guide. The piston has a considerable length of straight bore which pilots on the valve stem. The faces of both piston and nut and the barrel of the former are ground after hardening, the material being a chrome-vanadium case-hardening steel. Eight holes are drilled in the barrels of both the nut and the piston, to provide means for valve clearance adjustment.

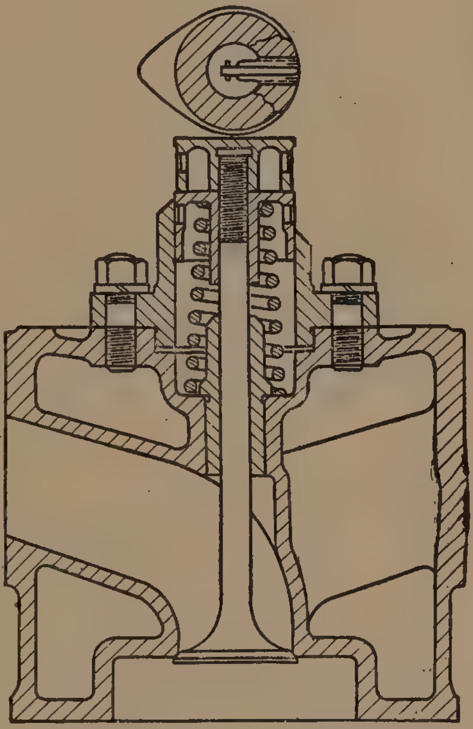


FIG. 211.—DIRECT-ACTUATED VALVE (STUTZ).

Overhead Camshaft and Rocker Levers—In most engines with overhead camshaft, the valves are operated through the intermediary of rocker levers, and an example of this method of valve actuation is shown in Fig. 212, which illustrates the cylinder head of the Italian Ansaldo engine, with its valves, camshaft and rocker levers. All of the curved rocker levers are threaded over a hollow spindle located directly above the camshaft. At their ends these levers are formed with hardened heels which contact with the cams and the valve stems. Adjustment of valve clearance in this case is made by means of the nut on the end of the valve stem, which is locked by the spring washer, the tapered end of the nut evidently being split, although this is not shown in the drawing. In other designs the arm of the lever which lifts the valve is provided with the usual adjusting screw, while in still others single armed levers are used, pivoted at one side of the cylinder head, carrying a roller at some distance from their free end, which serves as cam follower, and bearing against the valve stem with their outer end which is provided with an adjusting screw. An arrangement has been patented in England recently whereby valve clearance adjustment is effected by means of an eccentric bushing between the lever and the spindle around which it rocks, the bushing being split and provided with a clamp screw by means of which it can be clamped on the spindle in any angular position.

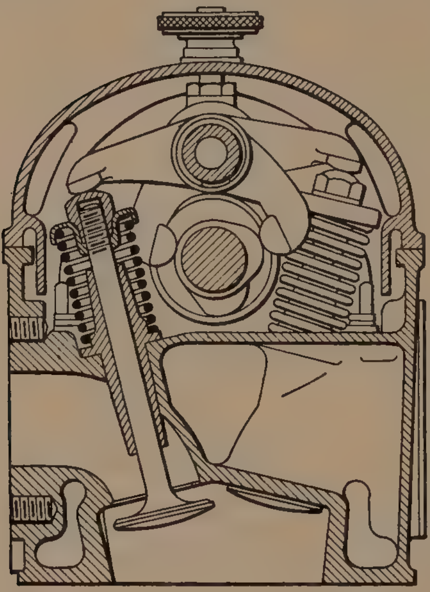


FIG. 212.—VALVES OPERATED FROM OVERHEAD CAMSHAFT THROUGH ROCKER LEVERS.

Compression Relief Cams—On this and the following pages will be described a number of cam and valve gear features which, though rarely used at the present time, should be known to the designer. One of these is the exhaust relief cam, which is of service particularly in facilitating the starting of large bore engines.

The exhaust cams are provided with an extra cam nose at one end and nearly opposite the main cam protuberance, which holds the exhaust valve open during the first part of the compression stroke. The camshaft is so arranged that it can be slid lengthwise in its bearings, and the cams are made considerably wider than the cam roller, so that when the camshaft is in its normal position the exhaust cam roller does not touch the relief cam. If the same camshaft carries the inlet cams, these latter also are made of extra width, so that the functioning of the inlet valves is not affected by the shifting of the shaft. The camshaft is connected to an operating lever from which connection is made to a handle or other operating device located at a point where it is conveniently reached by the driver when starting the engine.

A spring normally holds the camshaft in such a position that the relief cams are out of line with the cam rollers, but when the shaft is pulled forward by the operator the relief cams come in line with the cam rollers and the exhaust valves are held open not only during the regular exhaust period, but also during the first part of the compression period. The sides of the relief cam protuberances which the cam roller must mount are suitably inclined. A portion of an exhaust camshaft with relief cams and operating mechanism is shown in Fig. 212-A. At the forward end of the camshaft there is a short shaft, which is so connected to the camshaft that it does not need to turn with it, but that the camshaft must follow it whenever it is moved longitudinally. This short shaft is surrounded by a spring, which tends to keep the camshaft in its normal position. The forward end of the short shaft projecting through the cam gear housing cover is flattened and pivoted to an operating lever. The camshaft should be fitted with a ball thrust bearing to take up the end thrust due to the spring.

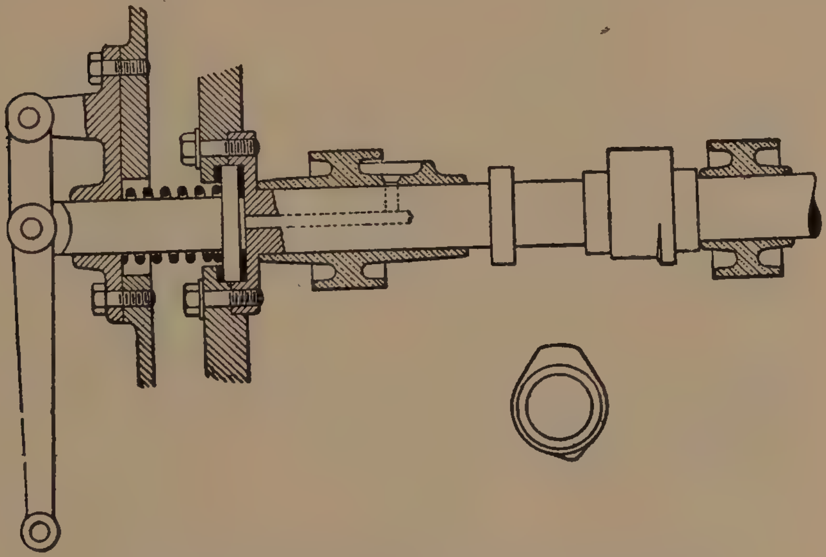
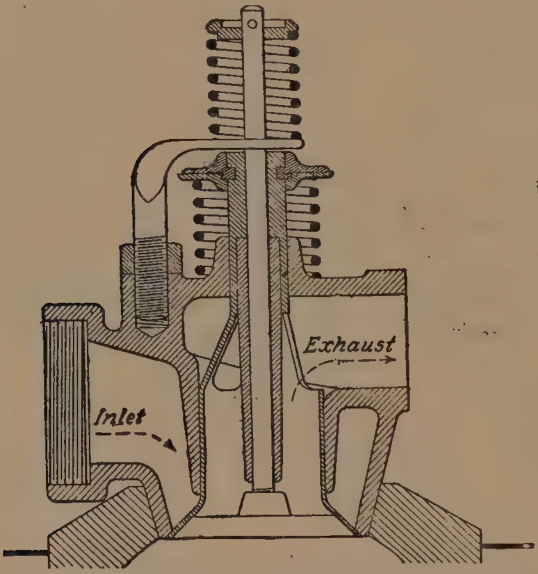


FIG. 212-A.—COMPRESSION-RELIEF MECHANISM.

Concentric Valves—Fig. 212-B shows a design of concentrically arranged inlet and exhaust valves as used by the H. H. Franklin Manufacturing Co. in its air-cooled engines for a number of years. At present no such valves are being used in automobile engines, so far as the author knows. One advantage of these valves is that they permit of large valve opening areas in connection with a combustion chamber of cylindrical or nearly hemispherical form, which is desirable from the standpoint of volumetric efficiency. Besides, the exhaust valve is cooled by the fresh charge coming in through the inlet valve, and is, therefore, not so easily pitted. The Franklin Company discarded these valves apparently because they proved noisier than ordinary valves.

Auxiliary Exhaust Ports—To insure the quickest possible escape of the exhaust gases, the cylinders are sometimes provided with ports, known as auxiliary exhaust ports, which are uncovered by the piston just before it reaches the bottom end of the stroke. These ports are opened very quickly and allow the greater portion of the spent gases to escape before the piston completes its down stroke. Before it was discovered that the exhaust valve had to begin to open some 45° to 50° before dead center in order to insure prompt clearing of the cylinder, and when one- or two-cylinder engines were the rule, these auxiliary exhaust ports were very extensively used, since it was found that if the exhaust valve opened only a little before the end of the stroke these ports considerably increased the engine power. However, when four-cylinder engines came into use, a difficulty cropped up. When the burning gases are discharged through the auxiliary exhaust port of one cylinder into an exhaust pipe common to all cylinders, they may flow back through the auxiliary exhaust port of one of the other cylinders whose piston is at the moment at the end of its inlet stroke and has therefore also uncovered its auxiliary exhaust port. In that case the charge in the latter cylinder may be fired, straining and heating the engine without doing any useful work. In order to prevent this it is



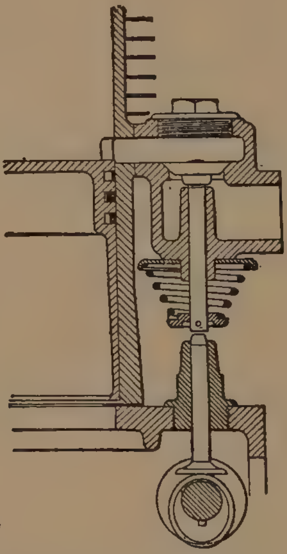


FIG. 212-B (LEFT).—FRANKLIN CONCENTRIC VALVES.
FIG. 212-C (RIGHT).—FRANKLIN AUXILIARY EXHAUST VALVE.

necessary to place a valve outside each of the auxiliary exhaust ports, which will shut off communication between the port and the common exhaust pipe at the end of the inlet stroke. (Fig. 212-C).

The last firm to use these auxiliary exhaust valves was the H. H. Franklin Mfg. Co., on an air-cooled engine. According to this manufacturer, only about 30 per cent of the total quantity of exhaust gas passed through the regular exhaust valves, which began to open only at the beginning of the exhaust stroke. These valves could, therefore, be made smaller than would otherwise be permissible; they did not need to be lifted against any pressure, and the strain on them and the consequent noise were much reduced. These advantages were obtained, however, at the cost of some complication, since an extra valve per cylinder was required.

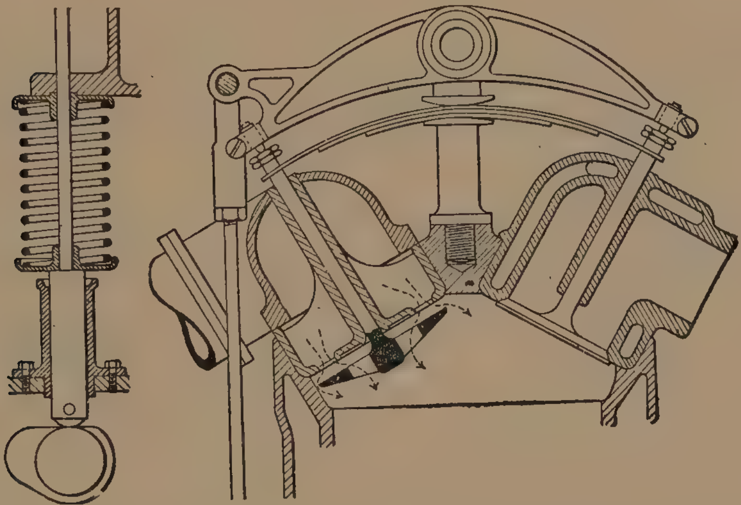


FIG. 212-D.—SINGLE CAM MECHANISM AND PORTED INLET VALVE.

Single-Cam Valve Gear—It is possible to operate both the inlet and the exhaust valve of each cylinder by means of a single cam, a construction which has been used on Fiat racing engines among others. The Fiat design is illustrated in Fig. 212-D. It will be seen that the two valves are located in the cylinder head at an angle of 30° with the cylinder axis and are held down to their seats by a multiple leaf spring. They are opened by means of a rocking lever pivoted to a bracket screwed into the center of the cylinder head, one end of which lever connects to a valve rod extending down the side of the cylinder and connecting with the cam follower at its lower end. This valve rod is surrounded by a coiled spring which is stronger than the flat spring holding the valves to their seats. It will be noticed that the cam has both a cam protuberance for opening the exhaust valve and a flat which allows the valve rod to descend below its normal position and the coiled spring to open the inlet valve against the pressure of the leaf spring.

Fig. 212-D also illustrates another idea in valve construction which has been used especially in racing and aircraft engines, in order to increase the power output. This is the ported inlet valve. It will be observed that the inlet valve head is provided with ports, so that the charge passing through the valve seat, instead of all passing around the outside of the valve head, comes partly around the circumference and partly through the ports in the valve head.

Dual Valves—Large valves are more troublesome than small ones, because a large disc will be warped more by the heat, and, besides, the weight of the valve increases rapidly with the linear dimensions, hence the strain on the valve due to rapid opening and closing becomes very great for large diameters. For this reason it has become customary in high speed engines with large cylinder bores to use two inlet and two exhaust valves per cylinder.

An experimental investigation of the relative capacities of large and small valves was made in connection with the development of the Liberty aircraft engine, and the conclusions reached were that at the same pressure drop, one valve of diameter D and lift h is equal in capacity to, first, a pair of valves of diameter $0.707D$ (equal port area) and lift $0.707h$ and, second, a pair of valves of diameter $0.6D$ and lift h, for values of h not exceeding about $0.25D$.

Most dual valve engines have been valve-in-head engines, but the T-head engine also lends itself to dual valve design. When in the cylinder head, the valves are preferably placed at a moderate angle to the cylinder axis.

Double Valve Springs—Double valve springs are often used in high speed overhead valve engines, the two springs being arranged concentrically around the valve stem. In a high speed engine very strong springs are required, and as the springs cannot be made very long, because it is necessary to limit the height of the engine, they must be loaded rather heavily, with the result that they are apt to break. Therefore, a second, less heavily loaded spring is placed inside the regular spring, so in case the main spring breaks, the operation of the engine is hardly affected, except at high speeds.

Dimensions of Camshaft—The camshaft must be comparatively stiff and must be well supported, so that it will spring or give very little under the shock it receives when opening the valves. This shock, of course, is greatest in the case of the exhaust valve, which must be lifted from its seat against the pressure of the gases in the cylinder near the end of the expansion stroke (about 50 lbs. per sq. in. gauge). To this must be added the initial spring pressure and the force required to accelerate the valve and push rod from rest. In the case of the inlet valve only the last two items need to be counted with, as the valve does not have to be lifted against a pressure.

For an accurate determination of the total thrust on the camshaft, the maximum force of the acceleration should be calculated from the cam profile, maximum engine speed, and weight of valve reciprocating parts. The thrust is particularly large with mushroom-type cam followers, which give the valve a very rapid acceleration. For an approximation it will be sufficient, however, to assume the total thrust on the exhaust cam under full engine load at the beginning of the lift to be 200 lbs. per sq. in. of valve port area with mushroom-type followers, and 125 lbs. per sq. in. with constant acceleration and tangential type cams. The load on inlet cams would be about 50 lbs. per sq. in. of valve port area less in each case.

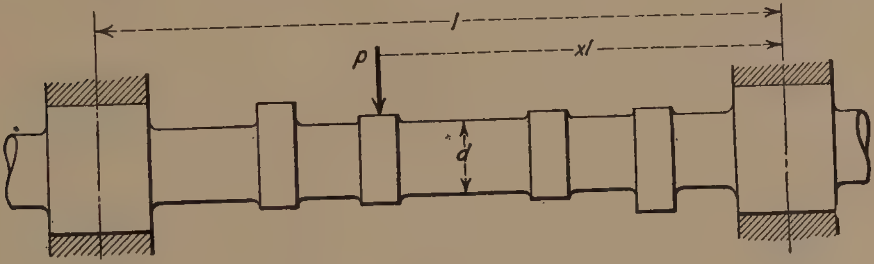


Fig. 213.—Camshaft-Load Diagram.

In Fig. 213 is shown a section of a camshaft of diameter d, whose length between the centers of adjacent bearings is l. Let the distance of the center of the cam from the farthest bearing be xl. Then the flexure of the shaft at the center of the cam under a total load P will be

$$y = \frac{Pl^3}{6EI}(2x^4 + 2x^2 - 4x^3).$$

(See Merriman, "A Textbook on the Mechanics of Materials"). The coefficient of elasticity may be taken at 30,-

000,000, and the moment of inertia I of a full circular section is $\pi d^4/64$. Inserting these values in the above equation we have

$$y = \frac{Pl^3}{8{,}800{,}000 d^4}(2x^4 + 2x^2 - 4x^3). \quad . \quad . \quad (86)$$

The deflection y should not exceed 0.003 in., and for engines that must operate quietly should preferably lie between 0.002 and 0.0025. For convenience in using equation (86) the values of the expression in parentheses ($= a$) for values of $x = \frac{1}{2}$ to 1, inclusive, are given in the following table:

$x = 0.5$	0.55	0.6	0.65	0.7
$a = 0.125$	0.122	0.115	0.104	0.088
$x = 0.75$	0.8	0.85	0.9	0.95
$a = 0.069$	0.051	0.033	0.016	0.005

Suppose that in a 4 × 5 in. engine there are three bearings for the camshaft, and that the distance between centers of bearings is 10 in. If all of the cams are on one side, there will be four cams between each pair of supports. Luckily the exhaust cams, on which comes the greatest thrust, will be closest to the bearings. The distance between the center of the bearing and the center of the nearest exhaust cam is 1½ in. and the distance between the center of the bearing and the center of the nearest inlet cam, 3¾ in. Hence the value of x is 0.85 for the exhaust and 0.625 for the inlet cam, making the values of a, 0.033 and 0.110, respectively. The area of the valve port for a 1¾ in. diameter is 2.4 sq. in. and the thrust on the cams therefore is

$2.4 \times 200 = 480$ lbs. for the exhaust, and
$2.4 \times 150 = 360$ lbs. for the inlet.

It is at once evident that, since the thrust on the inlet cam is three-fourths that on the exhaust cam but the moment factor a for the inlet cam is more than three times as great, the thrust on the inlet cam will cause the greatest deflection, and the inlet cam thrust therefore determines the necessary diameter of the camshaft. Making the permissible deflection 0.003 and substituting in equation (86) we get

$$0.003 = \frac{360 \times 10^3}{8{,}800{,}000 \times d^4} \times 0.11,$$

hence

$$d = \sqrt[4]{\frac{360 \times 0.11 \times 10^3}{0.003 \times 8{,}800{,}000}} = 1.10, \text{ say } 1\tfrac{1}{8} \text{ in.}$$

Camshaft Bearings—Camshafts located in the crankcase are usually supported by the same number of bearings as the crankshaft, these bearings being formed in the cross walls of the crankcase, the same as the crankshaft bearings. The bearing hubs or bosses are cast in, and when they are bored out the camshaft is inserted from the front end; consequently, all intermediate bearings must be of a diameter equal to slightly more than twice the maximum cam radius, so that the cams will clear the bearing surface. It is possible to make the bearings of the same diameter as the shaft itself, by using split bushings of considerable thickness, held together by machine screws, but this plan is no longer followed to any extent. A large "solid" bushing can be used for the front end, as it can be withdrawn with the shaft, and also at the rear end, as the rear bearing remains in place when the camshaft is removed.

The same as in the case of crankshaft bearings, camshaft bearings are often stepped down from front to rear, for convenience in boring them out. The length of the intermediate and rear bearings, if of the large diameter type, is made about one-half the diameter, while the length of the front bearing is made considerably greater and is usually determined by considerations of design. The front bearing, of course, has to carry the additional load due to the chain pull or tooth pressure. In the majority of engines the camshaft bearings are lined with plain bushings of either die-cast white metal or the bronze-back, babbitt-lined type, but in some of the lower priced engines the camshaft runs directly on the cast iron of the crankcase. For such service the cast iron forms a good bearing material, and the only disadvantage of the unbushed bearing is that it cannot be refitted.

Overhead camshafts are supported in pedestals mounted on the cylinder head, with removable caps, or else in a tunnel formed in the valve housing, into which they are inserted from the forward end, the same as in the case of camshafts in the crankcase. If carried on pedestals the bearing diameter is the same as the shaft diameter, and the bearings must then be made proportionately longer, to reduce the unit load due to the cam thrust.

Camshaft Manufacture—Camshafts are drop forged from low carbon case-hardening steel, and after being rough-turned, are cemented or carburized at the cam portions and then quenched from a temperature slightly above the critical point of the steel, so as to render the cam surfaces very hard and resistant to wear. To prevent carburization of

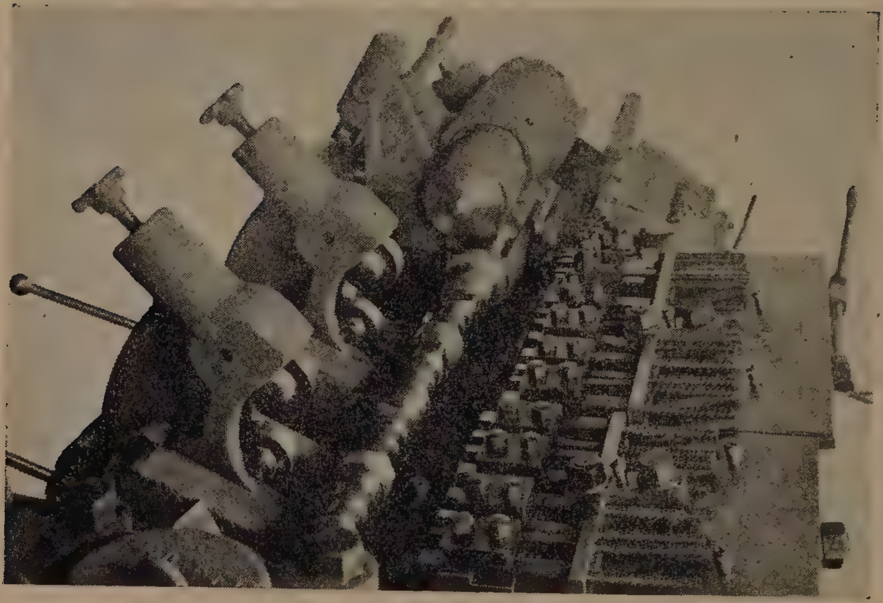


Fig. 214.—Tool Lay-out for Spacing Cams and Bearings.

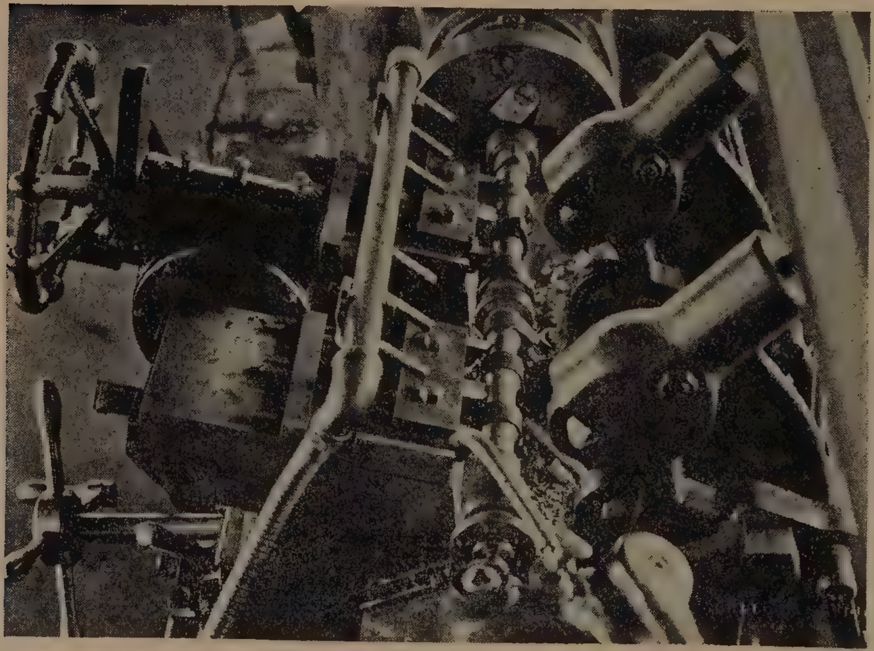


Fig. 215.—Tool Lay-out for Turning Camshaft in Lo-Swing Lathe.

the rest of the surfaces, the shaft is first copper-plated, and the layer of copper is then removed by machining where it is desired to carburize the surface. Carbon will not penetrate the layer of copper. Another plan consists in leaving those portions of the shaft which are to remain soft, oversize until after the carburizing process, and then turning off the carburized layer.

Machining Camshaft—All of the turning operations on camshafts with the exception of forming the contour of the cam are often performed in Lo-Swing lathes. With the proper tooling all of these surfaces on an average automobile camshaft can be turned in a minute and a half. In Fig. 214 is shown the attachment used with the Lo-Swing lathe for the spacing of the cams. With this attachment the sides of the cams and the bearings are all properly spaced and faced, and any chamfering or necking operations are also performed. As high as thirty-five and forty tools are used on a camshaft simultaneously. Of course, if all these tools were to act at once and all of them were to take the maximum cut, the shaft would be sprung so that it would be worthless. To overcome this, each slide carrying a set of tools, is fed into the work a little ahead of the slide next to it. In this manner the tools in the first slide, for instance, will be through their heaviest cutting by the time the tools in the second slide are ready to take their heavy cut, and so on to the third and fourth slide.

After the camshafts are spaced and faced with this attachment, they are put back in the lathe and the portion between the cams is turned. Generally this is done with simple tooling, although all bearings and spaces are turned simultaneously, thus reducing the time considerably. (Fig. 215.)

Cam-Grinding—A grinding machine attachment for grinding individual cams is illustrated in Fig. 216, and it may be pointed out that cam grinding machines designed to handle integral camshafts work on practically the same principle. The cam to be ground is secured on an arbor inserted into a spindle which carries the master cam at its other end. The spindle is carried in a head provided with an upwardly extending arm whereby it is swung from the body or frame of the attachment, which is clamped to the grinder frame. The swinging head is forced to one side by a coiled spring so that the master cam presses against a stationary shoe plate. The spindle carrying the cams is revolved by means of a work driving pin on the face plate of the machine engaging with the forked end of a driving arm on it. In order to make the necessary allowance for wear

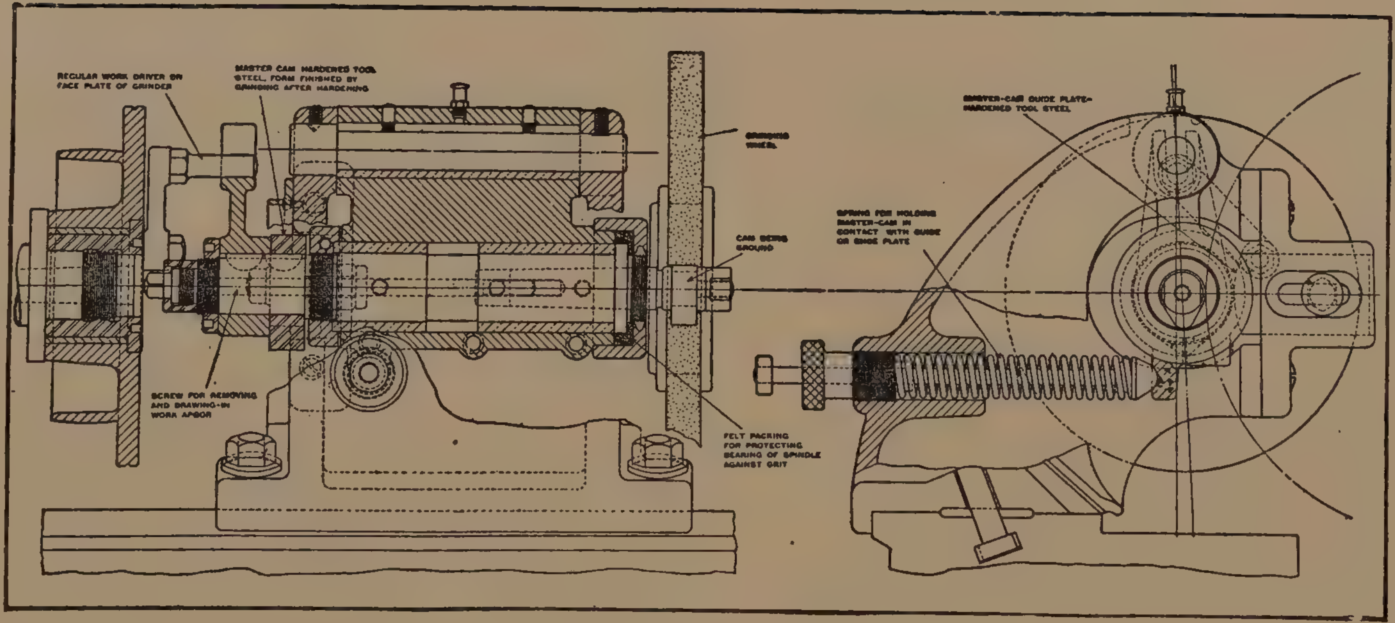

REGULAR WORK DRIVER ON FACE PLATE OF GRINDER
MASTER CAM HARDENED TOOL STEEL, FORM FINISHED BY GRINDING AFTER HARDENING
GRINDING WHEEL
CAM BEING GROUND
SPRING FOR HOLDING MASTER-CAM IN CONTACT WITH GUIDE OR GUIDE PLATE
MASTER-CAM GUIDE PLATE-HARDENED TOOL STEEL
SCREW FOR REMOVING AND DRAWING-IN WORK ARBOR
FELT PACKING FOR PROTECTING BEARING OF SPINDLE AGAINST GRIT

FIG. 216.—CAM-GRINDING ATTACHMENT (LANDIS).

in the grinding wheel, a set of different master cam shoe plates are furnished with the attachment, which are inserted one after another as the grinding wheel wears down, all of the plates having contact surfaces of different curvature. The master cam can also be cut by means of this attachment from a template. A method has been developed of grinding the cam with the flat sides of the grinding wheel, whereby inaccuracies due to change in curvature of the grinding surface with wear are eliminated.

For grinding an integral camshaft it is necessary to first make a master camshaft with all of the cams correctly spaced angularly and longitudinally. This camshaft is driven in synchronism with the camshaft to be ground, and acts on a roller carried by a cross slide carrying the grinding wheel, the roller being drawn against the master cam by a spring acting on the cross slide.

Two typical camshafts for six-cylinder engines, designed to be supported in three and four bearings, respectively, are shown in Fig. 217.

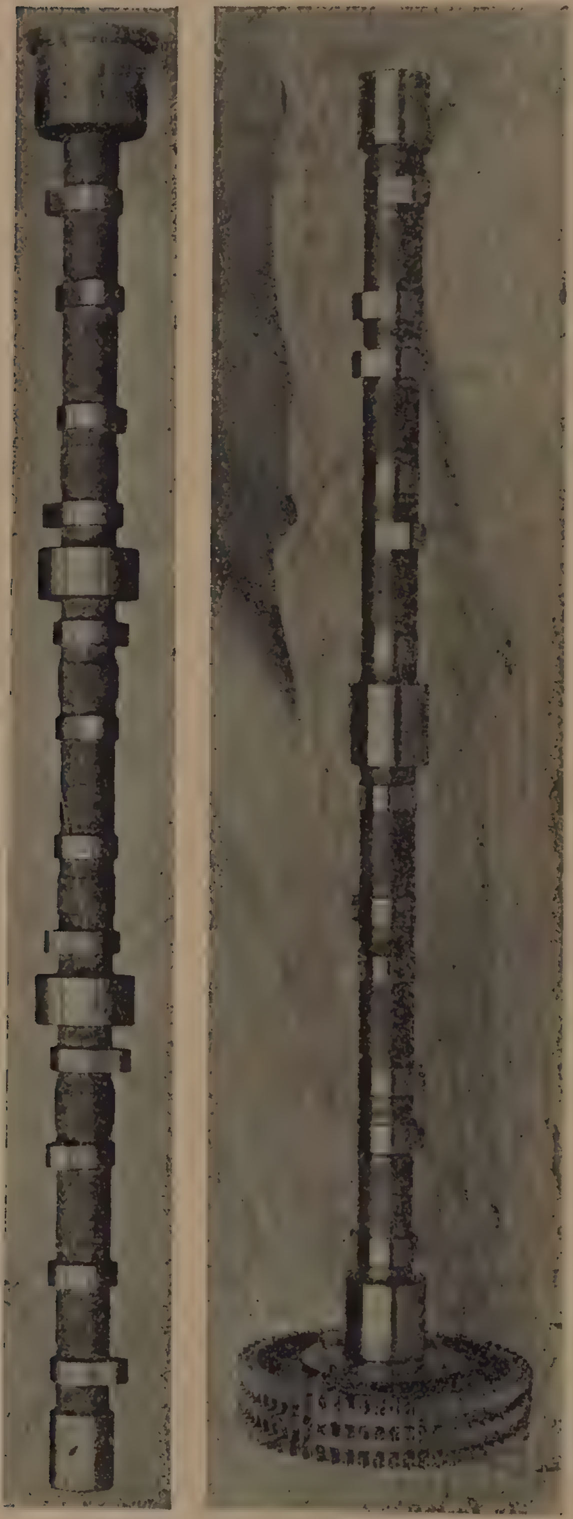


Fig. 217.—Two Designs of Camshaft for Six-Cylinder Engines with Three and Four Bearings.

CHAPTER XIII

Camshaft and Accessories Drives

The camshaft receives its motion from the crankshaft, from which all of the accessories must also be driven. In the case of passenger-car engines, these accessories usually include an oil pump, a water pump, a generator, an ignition unit, and a fan. On truck engines the generator is sometimes missing and the conventional ignition unit (interrupter and distributer) is replaced by a magneto. The camshaft and accessories drives are generally combined, all of the parts deriving their motion from a single pinion on the crankshaft, so it will be convenient to discuss the whole subject under one head.

In designing a drive for the camshaft and accessories of passenger-car engines, silence of operation is an important consideration. Most of the early automobile engines had a plain spur gear drive, but when Charles Y. Knight in 1905 introduced the sleeve valve engine to eliminate the noise of poppet valves, he found that in order to achieve any real improvement in the quietness of operation, he had to eliminate other sources of noise as well, and so for the half-time shaft he used a toothed chain drive.

With increasing engine speeds the problem of reducing gear noise became constantly more urgent, and from 1911 on the camshaft gears on all the more expensive cars that used a gear "front end drive" were cut with helical teeth, the pinion on the crankshaft at that time being made of steel and the gear on the camshaft of cast iron.

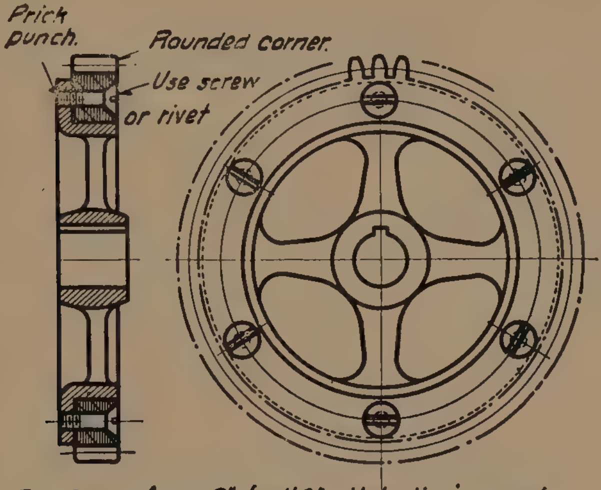
Camshaft Drive Practice—On truck engines the drive of the camshaft is almost always by means of metal spur gears cut with helical teeth, but on passenger cars camshaft drive practice varies. Of 111 models of passenger car produced in the United States in 1925, 60 had a camshaft drive by means of toothed chains, 38 by means of gear trains of which one member was made of non-metallic material, and 13 by means of all-metal gear trains. Of the 38 which used a non-metallic gear in the drive, 12 had the crankshaft pinion made of non-metallic material; 12, an idler gear between the crankshaft pinion and camshaft gear, and 14, the camshaft gear. The teeth of all gears in drives comprising one non-metallic member also are cut helically.

Location of Drive—Camshafts located in the crankcase (and overhead camshafts as well) may be driven from the crankshaft by gearing located at the front end, the middle or the rear end. In by far the majority of cases the gearing is located at the front end, because it is there more accessible than in any other place. There are, however, some very good reasons for placing the gearing either at the middle or at the rear of the engine, and this plan is followed by several European manufacturers. It was shown in the chapter on Balancing of Engines that the forward end of the crankshaft, particularly in the case of multi-cylinder engines, is apt to vibrate torsionally, and if such vibration occurs the camshaft driving gear will be subjected to stresses far greater than normal. To enable the ''front end drive'' to withstand these abnormal stresses, the gears or chains must be made far heavier than would be required to transmit the horse power absorbed by the camshaft if the driving torque were uniform. The flywheel has a steadying effect, and the closer we approach to it the more nearly uniform the speed and the driving torque become. Those who advocate the rear location for the drive assert that it is quite possible to make it outlast the engine in that location, so that the objection that it is inaccessible is of no moment.

Non-metallic Gears—About ten years ago the only non-metallic material used for gear wheels was rawhide, and this could not be used in engines, because when soaked with oil it swells and does not give the proper tooth contact. About 1915 a new class of non-metallic materials were introduced in the gear making art, composed of a base of cotton fabric impregnated with synthetic resins known as phenolic condensation products, of which Bakelite is the best-known. These materials have a considerable degree of elasticity, and while not quite as strong as cast iron under static load (a working stress of 6000 lbs. per sq. in. being recommended for low speed operation by some gear experts instead of the 8000 lbs. usually used for cast iron), it will withstand greater loads at high speed.

One of these non-metallic gear materials on the market is known as Bakelite Micarta and is manufactured by the Westinghouse Electric & Mfg. Co. It consists of a heavy duck of uniform weave, thickness, and tensile strength, bonded together with bakelite by heating while under high pressure. Besides its noiseless operation, the following advantages are claimed for this material. It is self-supporting, is not affected by water or oil, can be kept in storage indefinitely without deterioration, is not affected by most acid and alkaline solutions and has a long life.

Bakelite Micarta is manufacctured in plates or sheets approximately 36 in. square, and in thicknesses up to 2 in.; it is furnished either in plate form or cut into blanks, and for gears of large diameter it may be furnished molded into rings. Gear blanks are usually cut from plates by the gear manufacturers, the most efficient method of cutting the blank being by means of two trepanning tools, secured to a boring mill or drill press, the tools being so arranged that one will cut near the inside, and one near the outside, so as to prevent binding of the tools in the slot. When molded rings are desired, the treated duck is punched into rings or segments, which are placed in proper molds, and bakelized under pressure.



For Gears from 6" to 1'-0" pitch dia. inc and maximum width of 2"

FIG. 218.—NON-METALLIC CAMSHAFT GEAR.

Laboratory tests have shown the material to possess the following physical properties: Tensile strength parallel to laminations, 10,000 lbs. per sq. in.; compressive strength perpendicular to laminations, 35,000 lbs. per sq. in.; compressive strength parallel to laminations, 17,000 lbs. per sq. in.; transverse strength, maximum fiber stress both parallel and perpendicular to laminations, 17,000 lbs. per sq. in.; coefficient of expansion per in., per degree centigrade, 0.00002 in. in the direction parallel to the laminations, and 0.000085 in. in the direction perpendicular thereto. The specific gravity of the material is 1.4 and the weight 0.05 lb. per cu. in.

Micarta can be machined the same as metal. When cutting spur gears it is usually necessary to back up the blanks to prevent fraying of the material when the cutter comes through. This can be done economically by the use of hardwood blocks. A camshaft gear of Bakelite Micarta is shown in Fig. 218. This consists of a metal center with a gear ring made from plate material. At the present time non-metallic front-end-drive gears are generally molded of phenolic laminations, with a comparatively thin web, which, in conjunction with the inclined teeth, tends to cushion any shocks in the drive.

There are six or seven non-metallic gear materials of the phenolic condensation product type, and all have similar properties.

Arrangement of Gear Drives—Various arrangements of the camshaft and accessories drive by gears are illustrated in Fig. 219. At A is shown a simple drive, comprising a pinion on the crankshaft meshing directly with a gear of twice the number of teeth on the camshaft. With the latter gear meshes a smaller one for the pump and generator. This last-mentioned gear has the same number of teeth as the crankshaft pinion if the drive is for a four-cylinder engine, and two-thirds that number if for a six-cylinder.

With the drive shown at A, the generator and pump must be placed on the valve side, if the engine is of the L-head type. This is generally inconvenient, as the accessories would interfere to some extent with access to the valve adjustments, and the drive shown at A really is applicable only to engines with overhead valves with side rod operation. For L-head engines arrangement B is more suitable. Here the crankshaft pinion meshes with an idler gear, which in turn meshes with the camshaft at one side of the engine and with the generator and pump drive shaft on the other. In some passenger cars the accessories drive shaft is dispensed with by driving the generator by belt—sometimes the same belt that drives the fan—and the pump by direct connection to the camshaft, or else dispensing with it altogether by using the thermosiphon system of cooling water circulation.

Two drives suitable for truck engines are shown at C and D in Fig. 219. A magneto is generally standard equipment on the larger trucks, and while the maker does not furnish a generator with the truck, he wants to make provision so that the owner can readily have one installed if he so desires. Hence, drives for three accessories are required—magneto, pump and generator. These cannot well be all driven off one shaft, and at C, therefore, are shown two accessories drive shafts, the magneto drive shaft driven by gear from the camshaft, and the pump drive shaft similarly driven from the idler. If desired, a generator can be mounted in tandem with the pump and driven through its shaft.

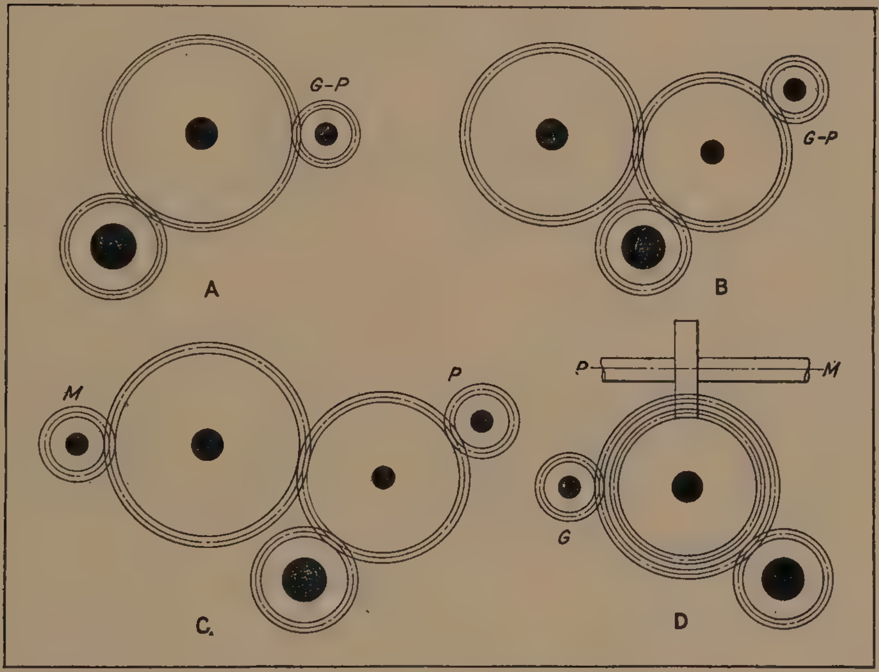


Fig. 219.— Arrangements of Gear Drives for Camshafts and Accessories Drive Shafts.

At D is shown a cross-shaft for driving the pump at one, and the magneto at the other end—an arrangement that has long been popular on the Continent. It has the merit of making parts that require occasional attention, such as the magneto interrupter and distributor and the pump packing glands, unusually accessible. This cross-shaft is driven from the camshaft through right-angled helical gears. The gear shown at G is generally not provided on the engine as the truck goes out from the factory, but provision is made for mounting a generator by making the front gear housing so it will accommodate a suitable gear and turning an opening in its rear side for mounting a generator by one of the S. A. E. standard mountings.

In nearly all front-end gear drives one member of the train is made of non-metallic material, but practice varies as to which one. One school of engineers say it should be the camshaft gear. This gear is large enough so it can be provided with a metallic center or hub, which can be accurately drilled and reamed and can have a key in it driven tight. There is no danger of the material being injured if in driving the key a blow should be misdirected. If a puller is to be used on the gear, the threads will hold much better in the metal. The frequency of contact of any tooth is less in the case of the camshaft than of the crankshaft gear, hence the material of least wear resistance should be used on the camshaft. The argument in favor of making the smaller gear non-metallic is that less of this expensive material is then required.

Proportions of Camshaft Gears—Since the camshaft must rotate at one-half crankshaft speed, its gear must have twice the number of teeth as the camshaft pinion. Either 8 or 10 diametral pitch is used, and in the case of helical gears the helix angle is made such that there is at all times pitchline contact at some point. In other words, the helical advance over the width of the gear must be equal to the circular pitch. For 10 diametral pitch the circular pitch is 0.314 and if the gear is 1 in. wide the tangent of the helix angle must be 0.314; that is the helix angle must be at least 17° 30′. For the same gear with 8 diametral pitch teeth the minimum helix angle should be 21° 40′. In general, if f is the face of the gear and d the diametral pitch, the helix angle should be such that

$$\tan \theta > \frac{3.1416}{p \times f}.$$

The face width with either cast iron or non-metallic gears in the train can be made

$$f = b/8 + \tfrac{1}{2}\text{ in.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (87)$$

where b is the bore of the cylinders. This may be increased by ⅛ in. in the case of truck and other heavy duty engines in services where weight is not an important factor. Sometimes the camshaft gears are cast with spokes, but the more common practice is to cast them with solid webs which may be machined down to a thickness equal to one-quarter the

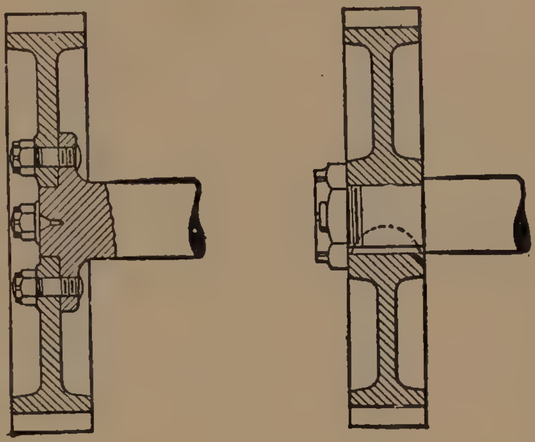


Fig. 220. — Fastening Camshaft Gears.

face width. Circular holes are sometimes cast in the webs to lighten the gears.

The crankshaft pinion is driven through a Woodruff key and is usually held against a shoulder on the crankshaft by spacers and a nut, which latter may form the starting crank ratchet. The preferred method of fastening the gear on the camshaft is by means of four cap screws which pass through holes drilled in the web of the gear and are screwed into a flange formed on the forward end of the camshaft. To lock them against coming loose, an iron wire is passed through a hole in the head of each cap screw and the ends of the wire are twisted together. The gear is piloted on the end of the camshaft. Of course, camshaft gears can also be secured to their shafts by a Woodruff key and nut, and this method also is in common use. The two methods of fastening them are illustrated in Fig. 220.

Gear Drives for Overhead Camshafts—Overhead camshafts of racing engines are sometimes driven by a train of spur gears. On passenger-car engines they are driven by either worm gearing or spiral bevel gearing. A worm gear drive is used in two English car engines, the Lanchester and Rover. In both of these there is a vertical shaft at the rear end, to one side of the central plane of the engine, which car-

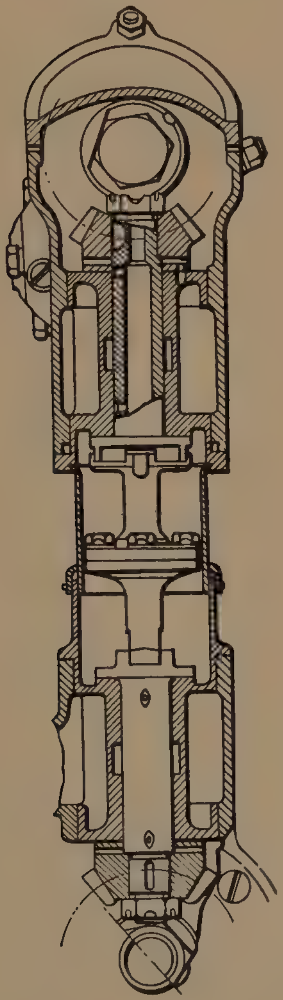


Fig. 221.—Gear Drive for Overhead Camshaft (Wills).

ries a worm wheel at its lower end and a worm or helical gear at its top. In the Lanchester the drive was formerly located at the front.

A drive by spur and bevel gears is used on the Wills-St. Claire engine (Fig. 221). This is a 60° eight-cylinder V engine, and at the forward end there is a two-to-one reduction by spur gears. The shaft carrying the driven gear extends centrally through the upper part of the crankcase and at the middle of the case carries a helical gear through which the oil pump is driven. In front of the driven gear on the half-speed shaft is mounted a spiral bevel pinion, which meshes with an equal sized spiral bevel gear for each of two inclined shafts. At its upper end each of these inclined shafts carries another spiral bevel pinion which meshes with an equal sized gear on one of the camshafts. The drive thus comprises seven steel spiral bevel gears, all of the same size.

In a drive of this type the two gears at the lower end of each inclined or vertical shaft must be carried in bearings which are supported by the same structural part or base, and whose relationship to each other therefore cannot change, and the same thing applies to the gears at the upper end of each shaft. The relationship between the crankcase and the cylinder head will change as a result of heat expansion of the cylinders, differences in thickness of gaskets, etc. To allow for such variations, and also to make possible the assembly of the drive in a housing without longitudinal joints, the inclined shafts of the Wills-St. Claire engine are made in four parts each. Of these the upper one, carrying the pinion of the upper pair of bevel gears, is rigidly supported in a bearing in the camshaft housing, while the lower one is similarly supported in a bearing fitting into the gear case cover at the front of the engine. The two intermediate parts of each shaft have flanges at one end by which they are bolted together, while their respective outer ends connect to the end sections of the shaft by driving couplings as shown, of a type which assures axial and radial freedom but allows very little angular play. The two inner sections of the shaft are surrounded by a housing in the form of telescoping tubes.

Chain Drive—More than half of all American passenger-car models at the present time employ toothed chains for the camshaft drive. These chains are made up of links stamped from steel plate, of the general form shown in Fig. 222, which in the original type were assembled by means of bushings and pins, the same as in roller chains; but considerable detail development work has been done on this type of chain and a number of notable improvements have been introduced.

The Morse Chain Company, in one type of chain known as the "rocker joint," uses a construction consisting of two hardened steel pins, one a rocker pin, the other a seat pin, passing through the holes in the links, the latter being held together by washers over the protruding ends of the seat pins, the pins being riveted at the ends. Three pitches of Morse chains are used for camshaft drives, 3/8, 4/10 and 1/2 in. Chains of the smallest pitch are of the pin and bushing construction. They are also adapted for the drive of single accessories, such as a generator or magneto.

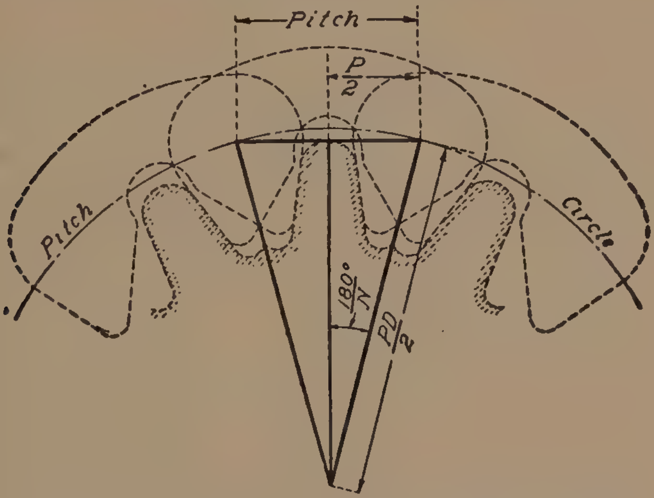


FIG. 222.—SPROCKET PITCH DIAMETER DIAGRAM.

Camshaft drive chains for passenger-car engines vary in width from 1 to 1¾ in., the 1¼ and 1½ in. widths being used most. For any given engine the life of the chain will vary with the width. Chain makers producing the open type of chain, contacting with only one-half the width of the sprocket tooth, recommend the use of a hunting link or offset link, the opposite ends of which are offset an amount equal to the width of the link. The effect of such a link is to transfer the contact from one half to the other half of the tooth surface for each round of the chain, thus uniformly distributing the wear over the whole width of the sprocket teeth. For guiding the chain on the sprockets, Morse makes use of a center guide link—a solid link at the center of the chain at alternate pitches—which runs with clearance in a groove in the sprocket teeth.

Calculations of Toothed Chain Center Distances—Where only two shafts are connected by a toothed chain the installation is sometimes made without provision for adjustment, and the necessary center distance of the shafts must then be calculated and the calculated dimension be closely adhered to.

Let P = pitch of chain and
N = number of teeth in sprocket,

Then

$$\text{Pitch diameter} = \frac{P}{\sin \frac{180^\circ}{N}} \text{ inches.} \quad . \quad . \quad . \quad (88)$$

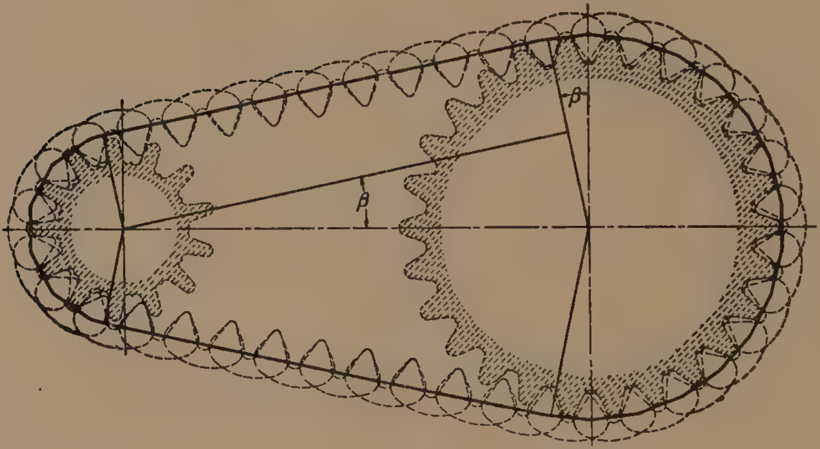


FIG. 223.—CENTER DISTANCE DIAGRAM.

Let Z = number of links in chain;
D = pitch diameter of sprocket wheel;
d = pitch diameter of sprocket pinion;
N = number of teeth in wheel;
n = number of teeth in pinion;
L = distance between axes of wheel and pinion;

Then

$$Z = \frac{N+n}{2} + (N-n)\frac{\beta}{180} + \frac{2D\cos\beta}{P}, \quad . \quad . \quad (89)$$

where

$$\sin\beta = \frac{D-d}{2L}.$$

The above formula can be used for determining the approximate number of links required for a certain center distance and sprockets with certain numbers of teeth. But since the result is almost sure to involve a fraction it cannot be used directly. The correct center distance for the nearest full number of links may then be found by means of the following equation:

$$L = \frac{P\left[Z - \frac{N}{2} - \frac{n}{2} - (N - n)\frac{\beta}{180}\right]}{2 \cos \beta} \quad . \quad . \quad . \quad (90)$$

The same principles as illustrated by Fig. 223 can be made use of in determining the length and number of links for a triangular drive.

Chain Adjustment—The preferred practice at the present time is to use triangular drives, the chain running over sprockets on the crankshaft, camshaft and accessories drive shaft, and in such drives means of adjustment are always provided. Adjustment is made either by varying the position of the axis of one of the sprockets relative to the other two, or else by providing an adjustable idler. According to one maker, the range of adjustment should be at least 10 per cent more than one pitch length.

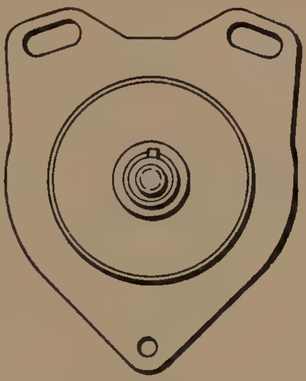


Fig. 224.—Generator Flange Mounting.

Chain adjustment can be effected in a very simple manner if the accessory to be driven, which is usually the electric generator, has an adjustable flange mounting on the chain housing. Referring to Fig. 224, the driving end of the generator is provided with a flange having three lugs for fastening it over an opening in the drive chain housing. The bottom lug has a hole of the size of the fastening cap screw, while the other two lugs have slots about 11/16 in. long through them, so the generator can be swung around the lower bolt as a pivot, for chain adjustment. The adjustment can be accurately made by means of a small set screw with lock nut, and after adjustment all three cap screws in the flange are tightened.

A spring drive for accessories has been developed by the Morse Chain Co. As shown in Fig. 225, the sprocket is mounted on a bracket with an extension for a bearing, and the armature shaft is driven from the sprocket through a laminated spring. With this mounting the generator can be removed without disturbing the sprocket. Lubrication is provided by a splash pocket in the bracket, with a groove in the bearing leading from same.

Morse Adjustment—A method of adjustment commonly used in camshaft drives furnished by the Morse Chain Co., is by an eccentric bushing through which the driven shaft passes and which carries the chain sprocket upon it. This adjustment is valuable where two or more accessories are connected in tandem. Referring to Fig. 226, the eccentric bushing forms an extension of a collar which is clamped or otherwise held against angular and axial motion in a bored hole in the rear wall of the chain housing. The bushing and collar have a common axis. If they are rotated around this axis the position of the driven shaft, which passes axially through the collar and bushing, is not changed, whereas the axis of the sprocket is shifted and the chain tension is adjusted. Owing to the eccentric relation between the sprocket and driven shaft, they cannot be rigidly connected together, and the drive is through a form of Oldham coupling.

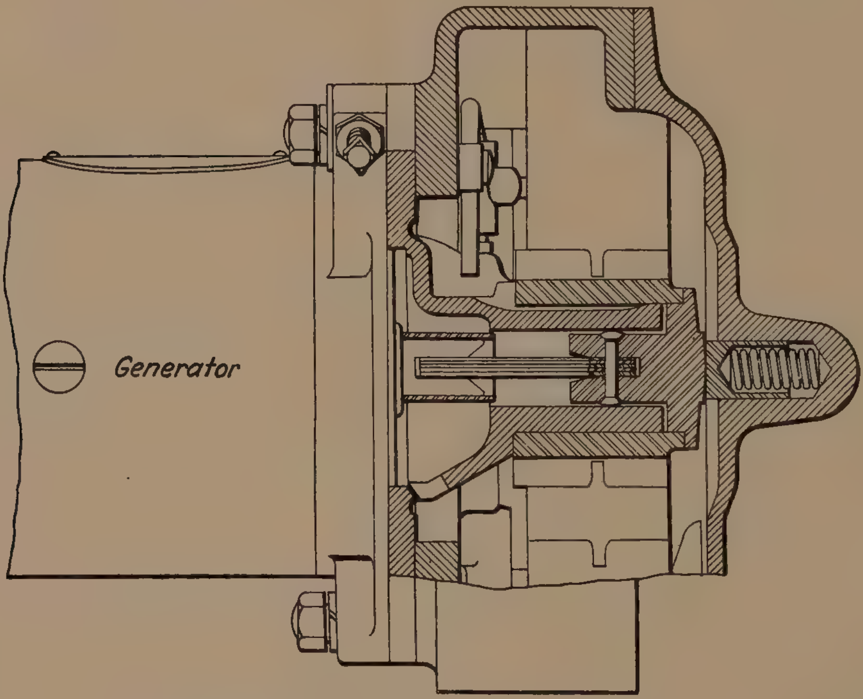
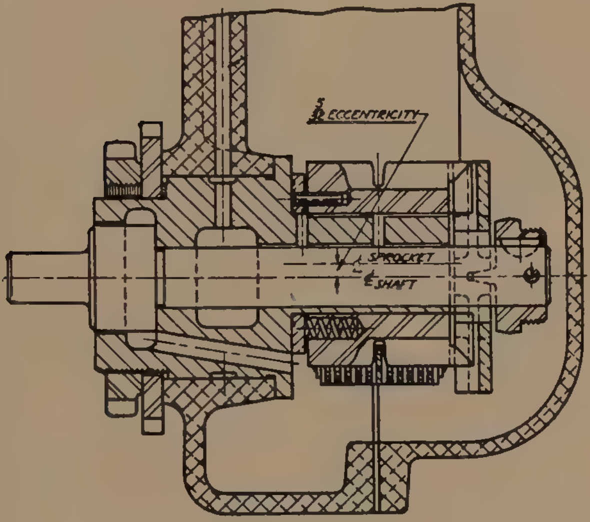


Fig. 225.—Morse Spring Drive.

Two methods of adjusting the collar and locking it in place are in use. One consists in cutting worm gear teeth in its circumference and mounting a worm meshing with it in the chain housing, which serves to rotate the collar around its axis, after which it is locked in position by pinching a flange on its outer end between the housing and a bracket flange bolted to the housing. The other method, illustrated in Fig. 226, consists in turning the collar with a wrench engaging into slots provided for the purpose, and then clamping it in the chain housing wall by a nut and locking the latter by a dog as shown. It will be seen that lubrication from the engine pressure oiling system is provided for.



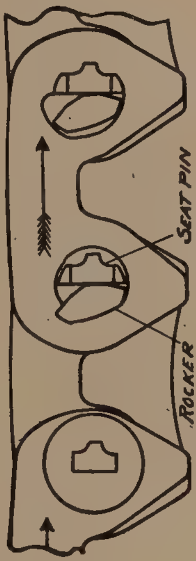


Fig. 226 (left).—Morse Chain Adjustment.
Fig. 226a (right).—Morse Rocker Joint Construction.

Rules for Chain Drive Design—Following are a number of hints on the design of toothed chain front end drives issued by the Morse Chain Co.:

In a triangular drive, whether the camshaft is on the right or left (looking at the engine from the front), the generator or accessories drive shaft must always be at the left of the camshaft and so located that the chain will not pull the camshaft over it. In laying out a triangular drive, the aim should be to get the maximum arc of chain contact (or wrap) on each of the sprockets, and in no case should the arc of contact be less than 95°. The length of chain should not exceed $39.5/p$ pitches, where p is the pitch.

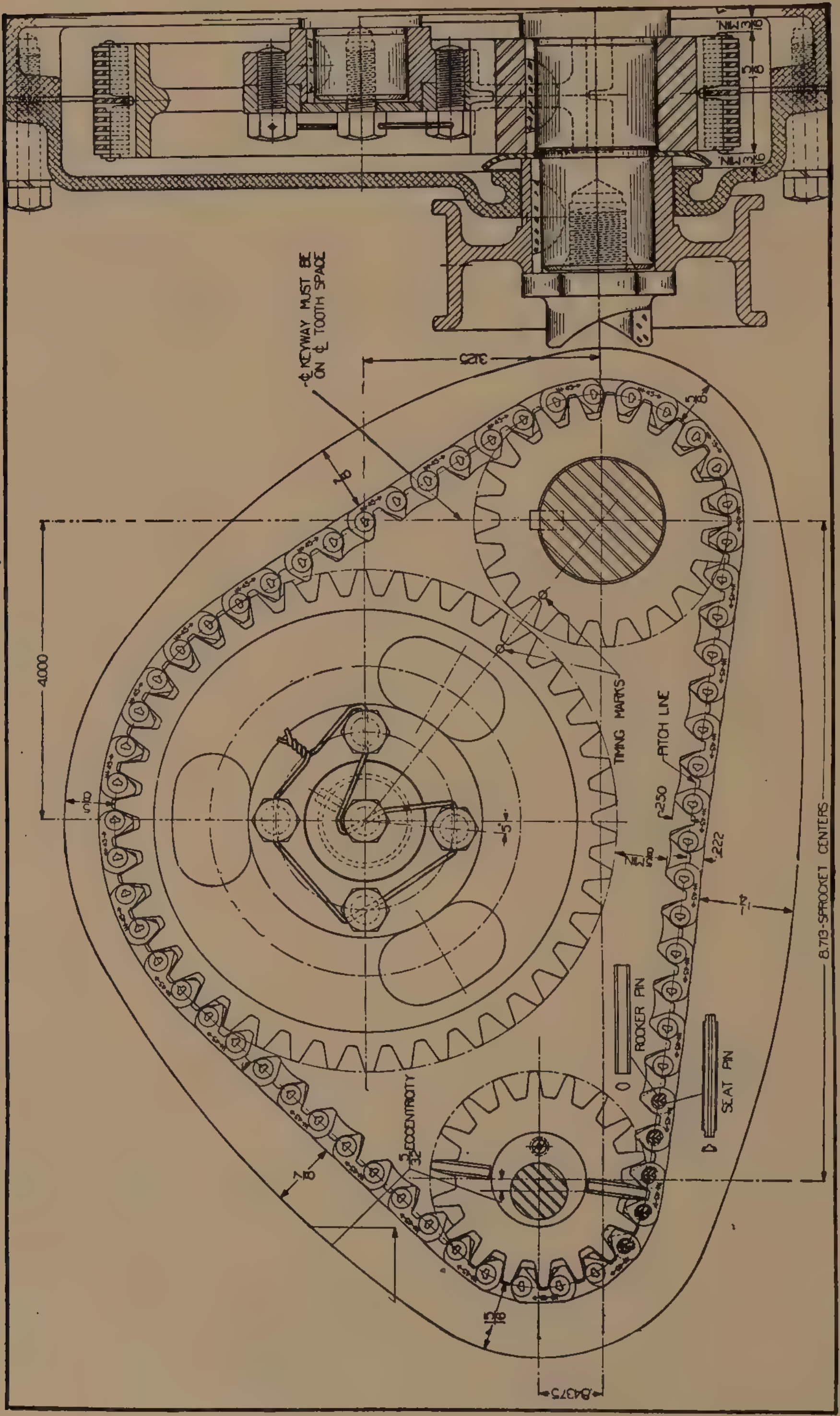


FIG. 227.—TRIANGULAR DRIVE BY MORSE CHAIN.

Sometimes, in cases where the above conditions cannot be met, it is possible to use a double drive or two loops of chain. In a simple drive from one shaft to another the drive-ratio must not be made too great; from the crankshaft to another shaft it should not exceed 1.25:1, and from the camshaft to another shaft it should not be greater than 2:1. In a two-sprocket drive the chain length should be not less than $16.5/p$ and not more than $32.5/p$ pitches.

For crankshaft sprockets best results are obtained with from $9/p$ to $10.5/p$ teeth, and the accessories drive shaft sprocket should be made with as many teeth as possible consistent with the speed requirements of the accessories to be driven. The fan may be driven by a separate chain from the camshaft, but it is better to drive it from the crankshaft, as the chain load is then reduced.

A typical triangular drive by Morse chain is shown in Fig. 227. There are 21 teeth in the crankshaft sprocket, 42 in the camshaft sprocket and 18 in the accessories drive sprocket. A ½-in. pitch chain is used, of 1½-in. nominal width and containing 63 links, of which one is a hunting link. Timing marks are placed on the crankshaft and camshaft sprockets, which for timing the drive are placed on the line connecting the axes of the two shafts. The camshaft sprocket is piloted on the end of the shaft and is secured to a flange on it by means of four cap screws, of which one is displaced 5° from the corner of the square marked by the axes of the other three, so as to make it impossible to incorrectly replace the sprocket on the shaft. Since the chain, on account of the use of a central guide link, must be placed on the sprocket from the top of the teeth, it is necessary to allow a clearance of at least ½ in. at the sprockets, and owing to variations in castings it is usual to allow ⅝ in. At the middle of the span this is increased to ⅞ in. in the case of the two short spans, and to 1¼ in. in the case of the long span, as indicated in the drawing. In the case of the accessories sprocket, 5/16 in. adjustment is provided for, and the clearance around it must be made correspondingly larger than at the fixed sprockets.

Link-Belt Chains—Two types of chain for camshaft drives are made by the Link-Belt Co., one known as the liner type, consisting of the usual assembly of flat links, while the other, known as the unit type, consists of special links as illustrated in Fig. 228. In Link-Belt drives adjustment of chain tension is made by means of an idler sprocket mounted on an eccentric stud in the chain case. The link of the unit chain is stamped with five lugs on one side, three at the outer contour and two on the outer sides of the punched holes. With the ordinary flat link chain, the pressure between the pin or bushing and the links, and the pressure between the links and the teeth are concentrated on one half of the available area of the bushings and teeth, respectively, and the unit pressures therefore are high. The object of the special form of link used in the unit chain is to distribute the pressures over the whole of the available area. With the larger area of contact, the oil cushion between the teeth of the chain and the sprocket, respectively, is more effective, and the life of the chain sprockets is increased, owing to the continuous nature of the bearing along the pin and the sprocket teeth.

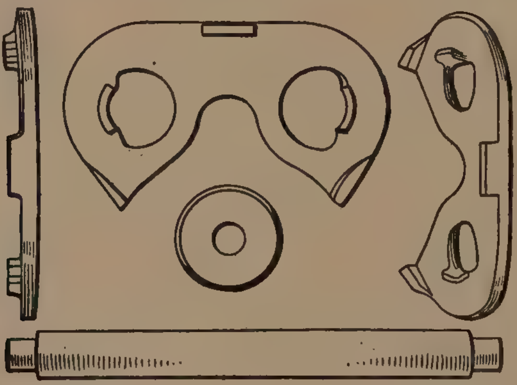


FIG. 228.—LINK-BELT UNIT-LINK, PIN AND WASHER.

With this type of chain no bushing is used, the links bearing directly on the pins.

The Link-Belt Co. also makes a double-sided chain, with flat links, in which some of the links are reversed, so they will engage with the teeth of an idler sprocket located on the outside of the chain. This form of drive is particularly advantageous in cases where the arc of contact would otherwise be insufficient. If the chain embraces a comparatively small arc there is danger of it jumping the sprocket after it has stretched at certain amount.

In Fig. 229 is illustrated the automatic chain-tensioning device used with Link-Belt drives. A stud A is clamped in a boss in the wall of the chain drive housing, and has mounted upon it the ecentric B, the right-hand end of which is enlarged to form a drum inside of which is located the clock spring D. The inner end of this spring is anchored to the collar E, which can be adjusted by means of the clamping screw on stud A, located outside the case, while the outer end engages into a slot in the drum. This device keeps the tension on the chain substantially constant irrespective of any "stretch" in service.

The Link-Belt front drive of the Locomobile is illustrated in Fig. 236. A 3/8-in. pitch chain is used, 1½ in. wide. The crankshaft sprocket has 28 teeth, the camshaft sprocket 56, the accessories sprocket 23 and the idler sprocket 35. The chain is of the double-sided type, running with its back side over the crankshaft sprocket. The use of an idler therefore makes it possible to have the accessories drive shaft to the right of the camshaft.

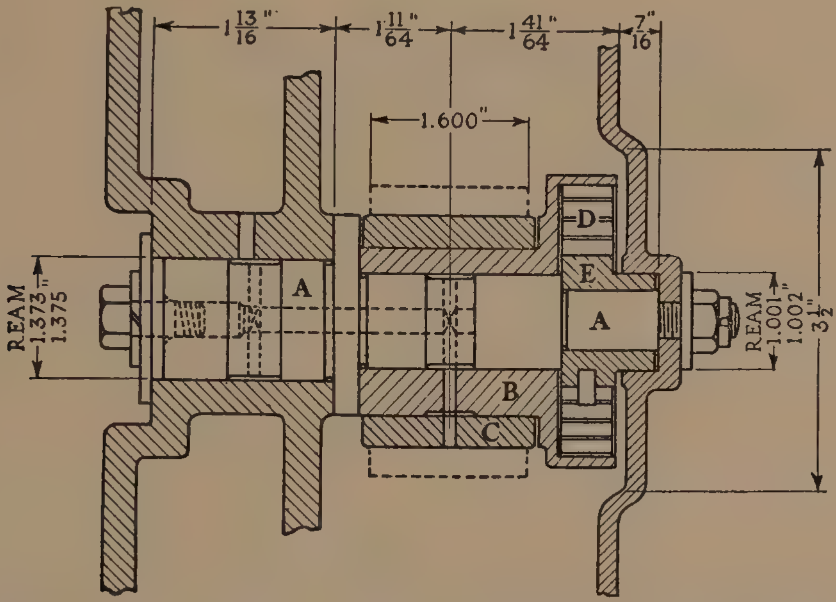


FIG. 229.—LINK-BELT AUTOMATIC CHAIN-TENSIONING DEVICE.

Double Chain Drive—On eight-cylinder V engines a double chain drive is sometimes used, the first chain driving from the crankshaft to the camshaft, and the second from the camshaft to the accessories shaft. This latter shaft generally drives the fan from its forward end and the generator, which is located inside the V of the engine, from its rear end. Such a double drive is illustrated in Fig. 230. Both drives being comparatively short, the chains are made wide and used without means for adjustment.

Generator and Pump Couplings—Both the generator and the water pump have their own shafts, which must be connected to the driving shaft or shafts by some form of coupling.

For driving the pump, a slightly flexible coupling is considered advantageous by some designers, for the reason that if any solid particles should accidentally get into the pump, a flexible drive would not be injured as easily, and if the pump should ever freeze up the flexible member might protect it against injury when the engine is turned over. In the case of both drives, it is well that the coupling allow of a slight disalignment of the two shafts connected, since it is very difficult to get these shafts absolutely in line with each other.

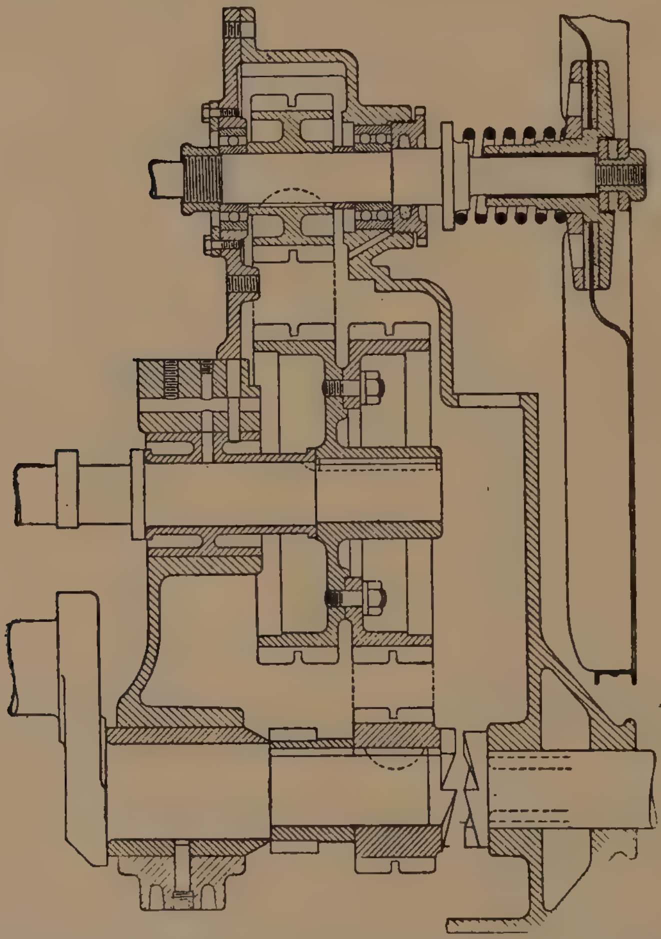


FIG. 230.—DOUBLE-LOOP CHAIN-DRIVE ON V ENGINE.

In Fig. 231 A shows the fabric type of coupling, which insures absolute silence of operation. It consists of two brackets at right angles to each other, secured by pins or keys to the driving and driven shafts, respectively, between which is located a circular disc of rubberized fabric, to which the brackets are secured by bolts. At B is shown the Bosch flexible coupling for magneto driving. This consists of a cylindrical steel block with a hub by which it is secured to the armature shaft. In the cylindrical block are cut two radial slots which are lined with fiber on their sides. To the driving shaft is secured a flange carrying a flat cross-bar, consisting of a large number of thin leaves of spring steel.

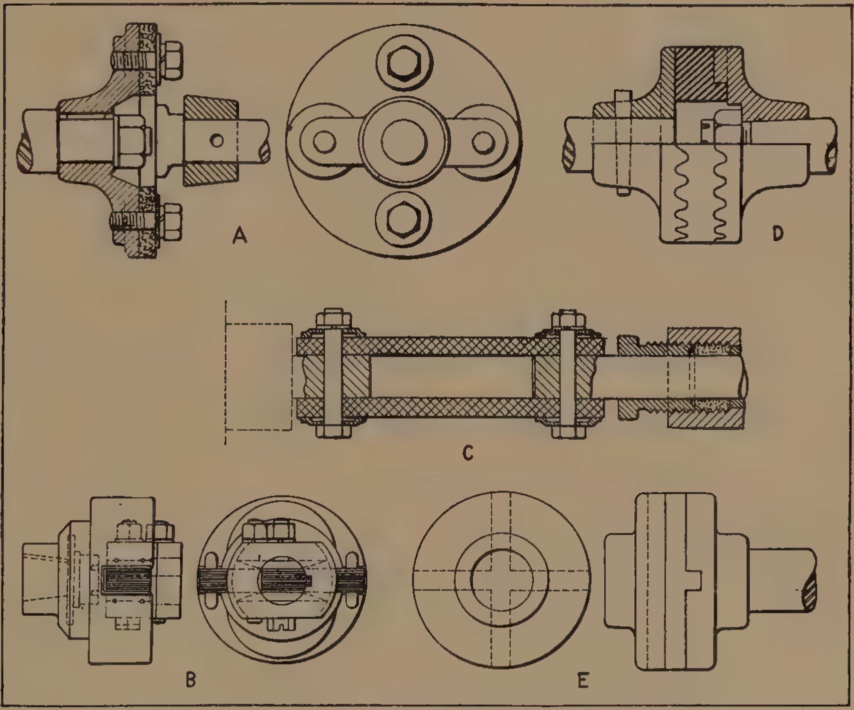


Fig. 231.—Accessories-Drive Couplings.

This cross-bar fits in the slots of the cylindrical member. It has a marked spring action, and provides for the slight relative torsional movement between the two members of the coupling resulting from the varying resistance of the magneto armature in the course of its rotation. It also has a slight bending or twisting motion sideways, thus permitting the two shafts to be slightly out of line.

The very simple generator drive coupling shown at C is used on the Star car. It consists merely of a length of rubber hose, slipped over the ends of the generator shaft and the drive shaft, respectively, and secured by clamping bolts through the hose and the shafts, with washers bent to conform to the circumference of the hose, under the head of the bolt and the nut. In this case the generator is placed near the rear end of the engine, and any misalignment of the shafts is readily compensated for by the flexibility of the hose. A similar drive is used for the water pump.

At D is shown a type of coupling which is much used on European cars. It consists of two metal discs keyed to the two shafts to be connected, the outer faces of which have teeth formed upon them similar to the teeth of a bevel gear adapted to mesh with a pinion of small diameter. Between these discs is located a molded rubber ring with teeth which exactly fit the spaces of the two metal rings. The drive thus is from one metal ring to the other through the rubber, the flexibility of which allows for any want of alignment and for eccentricity of the two shafts.

The Hookham joint type of coupling, shown at E, was much used in earlier years, but is less popular now, because it lacks flexibility. It consists essentially of three discs, the two outer ones being provided with hubs and keyed or otherwise secured to the driving and driven shafts, respectively. Adjacent faces of these discs are cut with diametral slots, and between the two discs is located a floating disc with tongues on opposite sides, at right angles to each other, which tongues are a sliding fit in the grooves in the outer discs. With this coupling, misalignment of the shafts causes no binding of the bearings.

Vernier Couplings—Occasionally couplings are used for driving the accessories which permit of accurately timing the driven part. One such coupling is shown in Fig. 232. Each of the two flanges is drilled with a series of closely spaced holes around its circumference, the circles of the centers of both series of holes being of the same diameter. One series contains two more holes than the other. The flanges are secured together by means of two bolts passed through holes at opposite ends of a diameter, and by moving these bolts from hole to hole the angular relation of the driving and driven parts can be changed in very small steps. Thus, if the one flange has 20 and the other 18 holes, the driven member can be advanced or retarded in steps of 2°.

Oil Pump and Ignition-Unit Drive—On passenger cars battery ignition is used almost exclusively, and the ignition unit combines an interrupter and a distributor which must be driven at camshaft speed. Sometimes provision is made for driving the ignition unit from the generator, the two forming, in fact, a complete assembly, so they can be installed

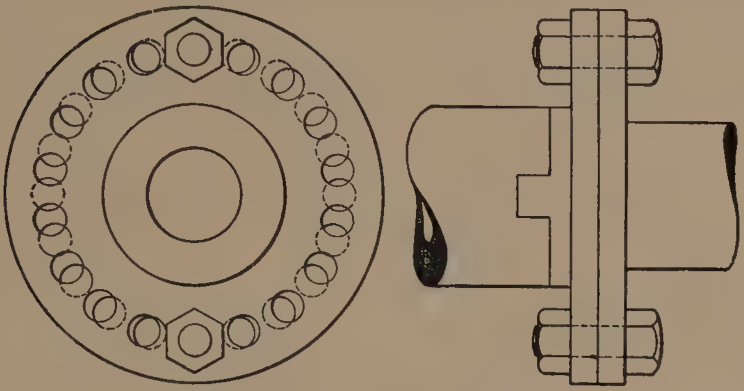


FIG. 232.—VERNIER COUPLING.

and removed together. Lately this practice is looked upon with less favor, because it makes operation of the engine dependent upon the presence of the generator.

At the present time a very common practice is to provide on the engine a shaft, either vertical or inclined, which is driven from the camshaft by helical gearing, generally located at the middle of the length of the engine, to the upper end of which shaft the ignition unit is secured, and to the lower end the oil pump. The drive at both ends is through a transverse integral key and a slot. This arrangement has

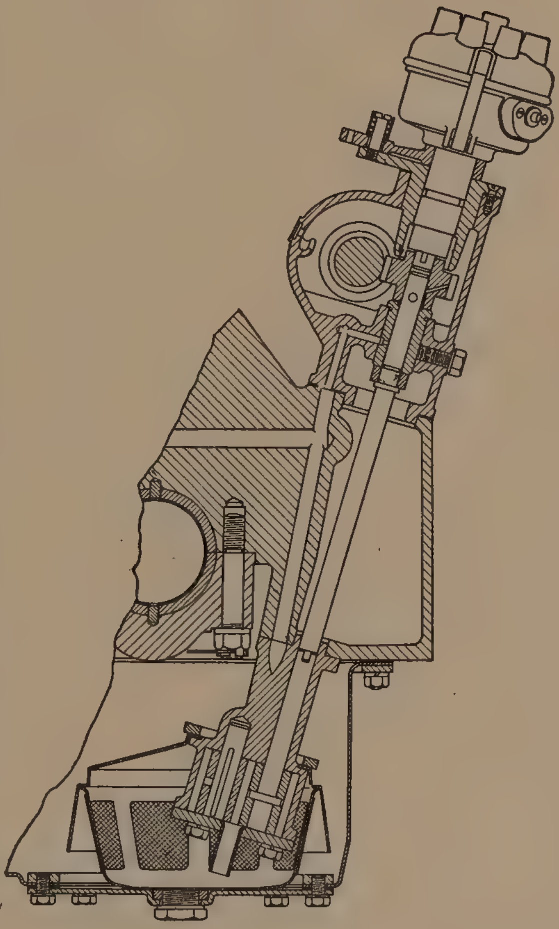


FIG. 233.—COMBINED OIL-PUMP AND IGNITION-UNIT DRIVE (HUPP).

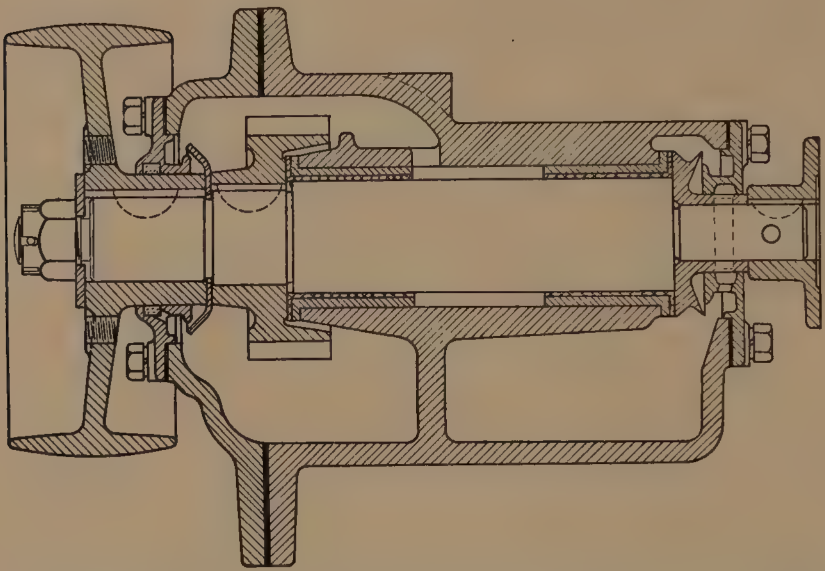


FIG. 234.—MOUNTING OF ACCESSORIES DRIVE SHAFT IN CRANKCASE.

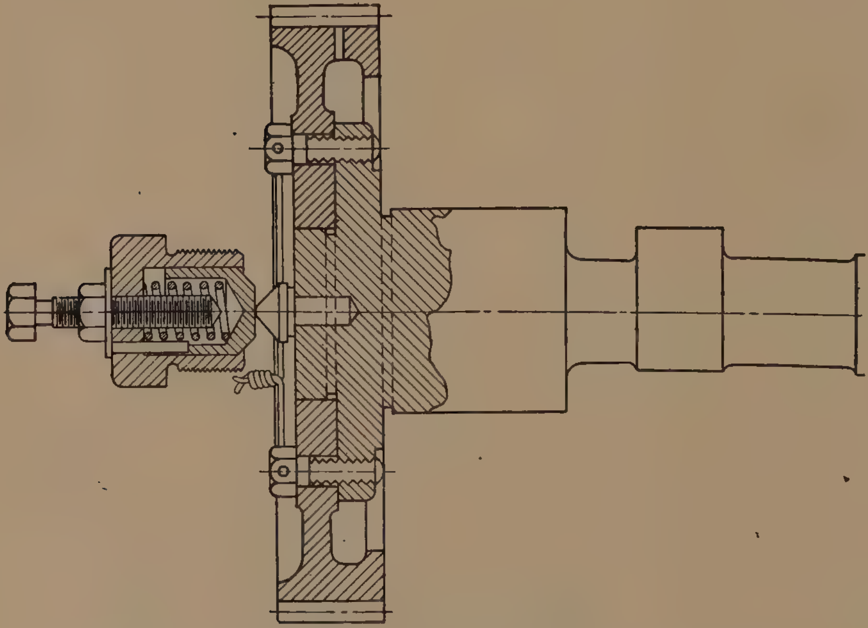


FIG. 235.—THRUST PLUNGER IN GEAR-HOUSING COVER.

the advantages that it makes one drive do for two accessories, places the gearing where it is well lubricated and well protected against dirt and grit, and brings the ignition unit into an accessible position at a minimum average distance from the spark plugs and the oil pump at the bottom of the crankcase, so it does not have to lift the oil by suction. The ignition unit and camshaft drive of the Hupmobile is shown in Fig. 233.

Fig. 234 shows a mounting for the water pump and fan drive shaft. The gear wheel indicated meshes with an idler gear in the camshaft drive and its shaft turns at crankshaft speed. Of the housing, the part on the right is cast integral with the crankcase, while that on the left is part of the cam gear cover. The housing being in communication with the timing gear housing, the bearing is oiled automatically, and oil is prevented from working out by knife-edge oil guards and packings in the end plates at both ends.

Camshaft Thrust Plunger—During the closing period of a valve the reaction between the cam follower and cam tends to cause the camshaft to run ahead of the crankshaft and to drive it, and if helical gears are used for the camshaft drive, there is, in consequence, a periodic reversal of the thrust on the camshaft which, if there is any appreciable endplay, may result in disagreeable noise. It is therefore well to take the play up by a spring-cushioned plunger as shown in Fig. 235. A hardened thrust button is inserted into a drill hole in the end of the shaft to contact with the thrust plunger. The thrust plunger is here shown screwed into the timing gear housing cover plate.

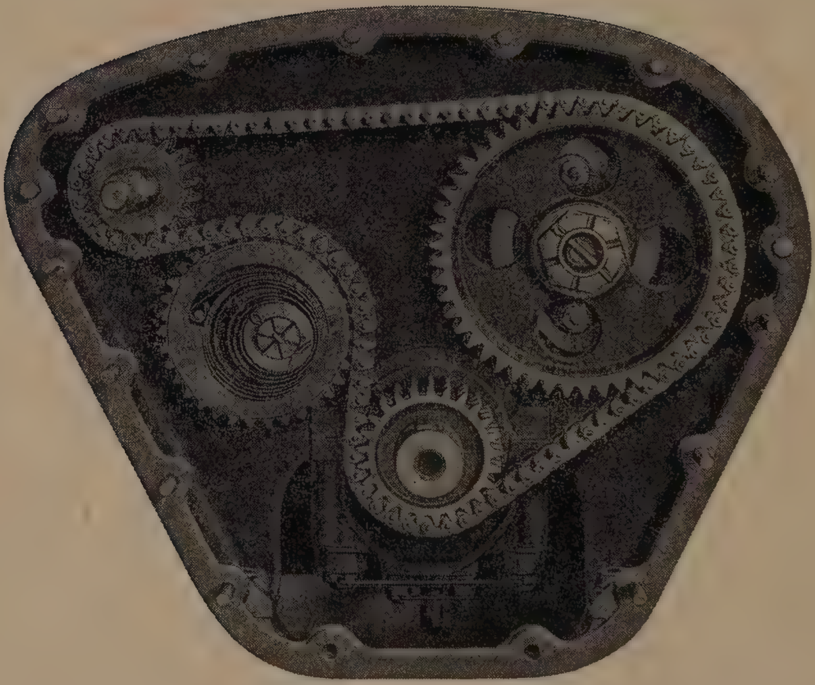


FIG. 236.—LINK-BELT DRIVE WITH AUTOMATIC TENSIONING IDLER. (LOCOMOBILE.)

CHAPTER XIV

The Crankcase

The crankcase in an automobile engine, if a separate member, serves as the main structural part which carries the cylinders and the crankshaft, and is in turn supported upon the vehicle frame. It also forms a housing for most of the important working parts of the engine, protecting them against dust and splashing mud, and it performs important functions in connection with engine lubrication.

The crankcase generally forms a box-like housing sufficiently wide at the crank bearings to enable the crankshaft with the connecting rod heads to rotate freely within it, and sufficiently long to accommodate all of the cylinders of the engine. It is provided with circular openings over which the cylinder casting is bolted, with bearings for the crankshaft and camshaft, and with supporting arms by means of which it is carried on the main frame or a subframe.

Formerly crankcases were almost always separate castings, but the practice introduced by the Ford Motor Co. about 1909, of casting all of the cylinders and the upper section of the crankcase in one unit, has been largely followed by others in recent years. It reduces the cost of manufacture and gives a more rigid engine structure, and the only objection to it is that it necessitates making the crankcase of the same material as the cylinder block, viz., cast iron, when, if it were a separate casting it would be made of a light alloy and a considerable amount of weight saved.

Materials—Crankcases of heavy truck and tractor engines are cast of iron and occasionally of semi-steel, also known as high tensile cast iron. Those for passenger-car engines, on the other hand, are usually cast of an aluminum alloy, in order to reduce their weight. The alloy most commonly used in this country is the No. 12 or S. A. E. specification No. 30. This contains a minimum of 90 per cent aluminum, 7-8.5 per cent copper, not over 0.20 per cent zinc, not over 1.70 per cent of silicon, iron, zinc, manganese and tin combined, and no other impurities. A test specimen of this alloy of ½ in. diameter, cast in sand and tested without having the skin removed by machining, will show from 18,000 to 20,000 lbs. tensile strength and 1 to 2 per cent elongation in 2 in. The specific gravity of this alloy is 2.83. A shrinkage allowance of 5/32 in. per foot should be made in patterns.

In England an aluminum-copper-zinc alloy is widely used for crankcase castings. It contains a minimum of 81 per cent of aluminum; 2.25-3.25 per cent of copper; 12.50-14.50 per cent of zinc and a maximum of 1.70 per cent of silicon, iron, manganese and tin, with no other impurities.

The tensile strength of this alloy when tested under the conditions outlined above, is between 25,000 and 30,000 lbs. per sq. in., and the elongation should exceed 1 per cent in 2 in. The specific gravity is about 3. Shrinkage allowance is the same as for the No. 12.

Types of Crankcase—Crankcases are usually divided horizontally into an upper and a lower section. The plane of the joint between the two sections may be either on the level of the crankshaft axis or lower. In all engines of large production the lower section is made of sheet steel and is formed in presses. If the plane of division is at the level of the crankshaft axis, more of the crankcase wall will be of sheet metal, and if the upper section is of cast iron, this will obviously reduce the total weight. On the other hand, if the upper section extends below that axis the whole engine will be of greater structural rigidity. Stiffening the crankcase in this way is the more important if the crankshaft has several throws between supports and there are, in consequence, only few cross walls. The predominant practice in the United States is to carry the upper section from 2 to 3 inches below the crankshaft axis, though in some cases practically the whole of the crankcase is in a single casting and the lower section is merely an oil pan.

Crankcases have been made also without any division in the horizontal plane. These are known as barrel type crankcases. At least one end wall is made in the form of a flanged and bolted circular plate, which covers an opening through which the crankshaft is introduced. This type is now practically obsolete, however.

The cylinders, of course, must be securely fastened down to the crankcase. The force of the explosion, pressing against the cylinder head, tends to lift the cylinders from the crankcase and to strip the threads of the cylinder retaining bolts. The problem of giving sufficient strength to the cylinder bottom flange and the retaining screws is of considerable importance where single cylinders are used, but in the case of multiple cylinder blocks it is easily solved, as the individual explosions are smaller and the resulting stress is divided among a greater number of bolts. There is usually one retaining screw or stud between adjacent cylinders on each side, and in addition there is one on each outer corner of the base flange at the end cylinders.

Aluminum does not hold threads nearly as well as cast iron, and as long as only single- and twin-cylinders of comparatively large bore were used in automotive practice it was considered unsafe to screw the retaining bolts or studs into the aluminum and trust to the threads therein. At the present time, with four-, six- and eight-cylinder blocks, the same fastening methods are used as with cast iron crankcases, except that the thread for the studs in the aluminum is made from 2 to 3 diameters long, instead of 1 to 1½ diameters, as in cast iron.

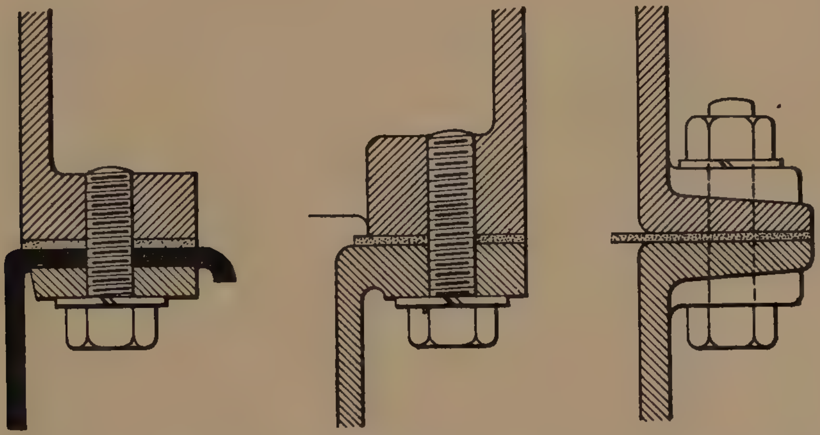


Fig. 237.—Types of Joint between Upper and Lower Sections of Case.

Where it is considered unsafe to depend upon threads in the aluminum, one or the other of the fastening methods shown in Fig. 238 may be used.

Thickness of Sections—The outer walls of a cast iron crankcase vary in thickness from 3/16 in. in the case of engines of less than 3-in. bore, to ¼ in. in engines of 4½-in. bore or over. Cross walls are usually single and re-enforced by suitable ribs. If the crankcase is divided through the crankshaft axis, there will be a horizontal rib from the bearing to each side wall, the lower edge of which will be slightly above the crankshaft axis, so it will not interfere with the milling operating on the bottom of the case. In addition there may be vertical or inclined ribs. In many cases there is an enlargement in the cross wall extending from the main bearing to the camshaft bearing, which is drilled out to form an oil duct furnishing lubricant to the camshaft bearing.

At the bottom the crankcase casting is provided with an outside flange which is made somewhat different in form according to whether the lower half of the case is a casting or a stamping. If the former, the lower half adds to the transverse rigidity and the flange need not be made quite so large. The minimum width of the finished surface at the bottom of the case is about 1 in., while in larger engines it is made 1¼ in., but beyond this finished portion of the flange there is sometimes an extension of ⅜ to ⅝ in. in the width of the flange for additional transverse stiffness. This is raised slightly above the rest so it need not be machined. Either 5/16 or ⅜ in. bolts or studs are used for joining the lower to the upper section of the case, and these are placed at from 3 to 3½ in. center distance. Three types of joint between the upper and lower sections of the case are illustrated in Fig. 237. A gasket of cork or tough, heavy paper is placed between the upper and lower sections. The construction shown in the center has the advantage that the outside of the crankcase, as seen from above, is quite smooth and can be easily kept clean.

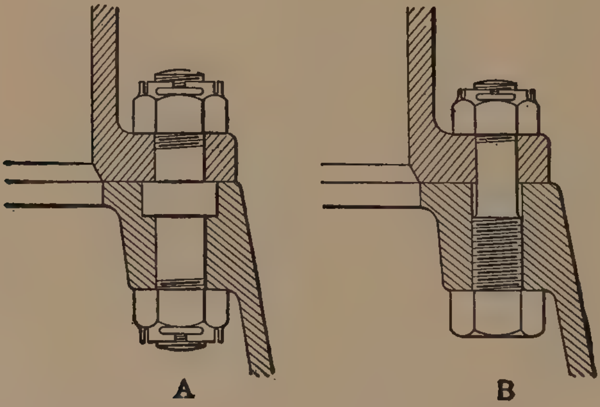


Fig. 238.—Cylinder Hold-Down Bolts Not Dependent upon Threads in Aluminum.

All of the crankshaft and camshaft bearings are supported in cross walls of the crankcase. At the top of a crankcase for a vertical engine there is a plate which has a number of circular openings formed in it corresponding to the cylinders. The cylinder block is bolted to this part of the crankcase, which is therefore made of somewhat greater thickness than the rest of the walls. The thickness of this top wall varies from 5/16 in. for an engine of less than 3-in. bore to ½ in. for a truck engine of 4½ to 5-in. bore. Where the cylinder retaining studs or screws come, bosses are provided on the under side of this wall, which in cast iron crankcases vary in depth from ⅜ in. for engines of less than 3-in. bore to ¾ in. for engines of 4½ to 5-in. bore. If there is no main bearing between a pair of adjacent cylinders, and consequently no cross-wall in the crankcase, a rib is run across the case underneath the top plate, the depth of which varies from ⅜ in. in small bore to 1 in. in the case of large bore engines. These cross ribs extend through bosses for the cylinder retaining bolts located on opposite sides.

General Form of Crankcase—Since the minimum cross-section of the crankcase depends upon the sweep of the connecting rod head, one might expect the crankcases to be made substantially cylindrical in shape. Such crankcases were in vogue at one time, but at present the majority are of more nearly rectangular section. In an L-head engine, on the valve side the top wall is carried out as far as the valve chamber extends out from the cylinder block, to form the floor of that chamber, and the side wall of the crankcase on that side is usually plane and substantially vertical. On the opposite side the upper half of the wall is usually substantially cylindrical, unless the water jacket extends all the way down to the bottom of the cylinder and is continued in the crankcase wall, in which case the latter also may be nearly plane and vertical.

The lower section or oil sump, which is made of cast iron in the case of most heavy truck engines, of cast aluminum in the case of passenger-car engines of moderate production, and of pressed steel in the case of large production passenger-car engines, is usually of trough-like form. In many engines the deepest section of the sump, and, therefore, of the whole crankcase, is at the middle, in which case the intake of the oil pump and its strainer are located there. In other cases the sump is of nearly uniform depth or deepest at the rear, and the oil pump inlet is located near the rear end. It is a good plan to provide the oil sump with two cross walls or partitions so the oil in it will not all run to one end when the car is on a steep grade. If the oil intake is at the rear end, then it will always be submerged in oil when the car ascends a grade; it may be out of the oil when running down hill, but that will be of little consequence.

Cross Walls or Bulkheads—Usually the partition walls which carry the bearings are single walls with ribs on both sides. There is one rib at the bottom of the bulkhead, which will be horizontal if the crankcase is divided through the crankshaft axis, and inclined from the main bearings down if the plane of division is below the crankshaft axis. Then there are often additional ribs running from the sides of the bearing straight up to the top wall and possibly also at an angle to the side walls. Directly above the bosses for the bearing cap bolts openings are cored in the bulkhead so that the drill and tap may run out. Occasionally double bulkheads, or what are sometimes referred to as box girders, are used.

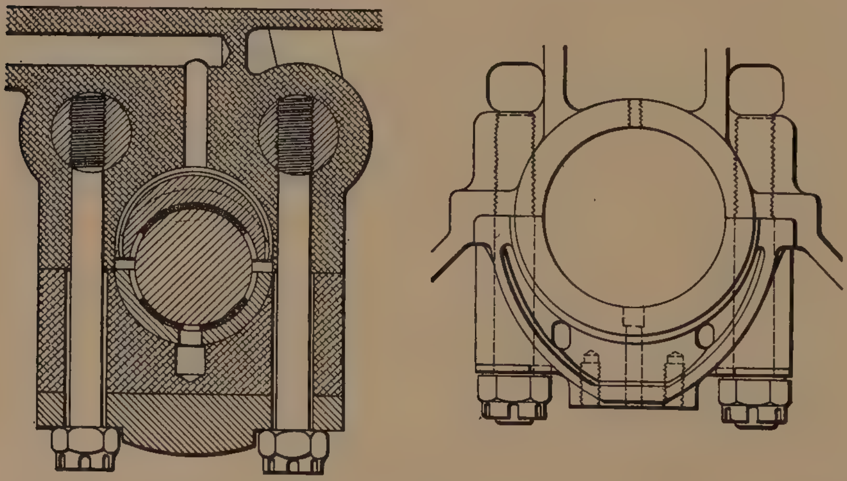


FIG. 239.—ARRANGEMENTS OF CRANKSHAFT BEARINGS AND CAPS.

Bearing Caps—The bearing caps are preferably made of the same material as the crankcase itself, as this obviates trouble in boring out and reaming the bearings due to unequal cutting qualities of the halves. Caps of bearings generally are held on by two studs and nuts each, those of wider ones by four. From a bearing load diagram, constructed as explained in Chapter X, the load on the studs can be determined and their required size calculated. The caps themselves must be sufficiently rigid so they will not deform appreciably under the varying loads on the bearing, and a rib is usually placed at the center as shown in Fig. 239. In the case of aluminum caps, since the resistance of aluminum to crushing stresses is not very great, some engineers plane or mill the bottom of the cap off flat and place a steel plate under it, on which the nuts and washers bear.

In order to secure accurate registry of the caps with the upper halves of the bearing, the former are recessed into the crankcase. Bosses for the retaining studs are formed on the caps and in the bulkheads. In a few engines the bearing caps are held on by "through" bolts extending all the way through the crankcase, cylinder block and cylinder head. In one respect this construction has become more practical with the increase in crankshaft diameters, as this brings the cap bolts farther out from the center plane of the engine. In a four-cylinder engine built by Rover in England cylindrical blocks of steel are pressed into corresponding longitudinal holes in the bulkheads and drilled and threaded to receive the bearing cap bolts, as illustrated in Fig. 239, to obviate dependence upon threads in aluminum.

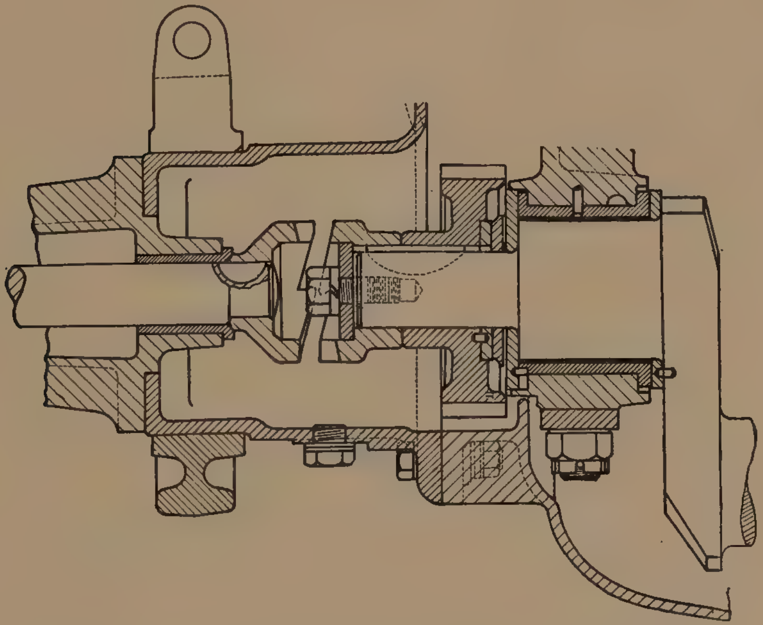
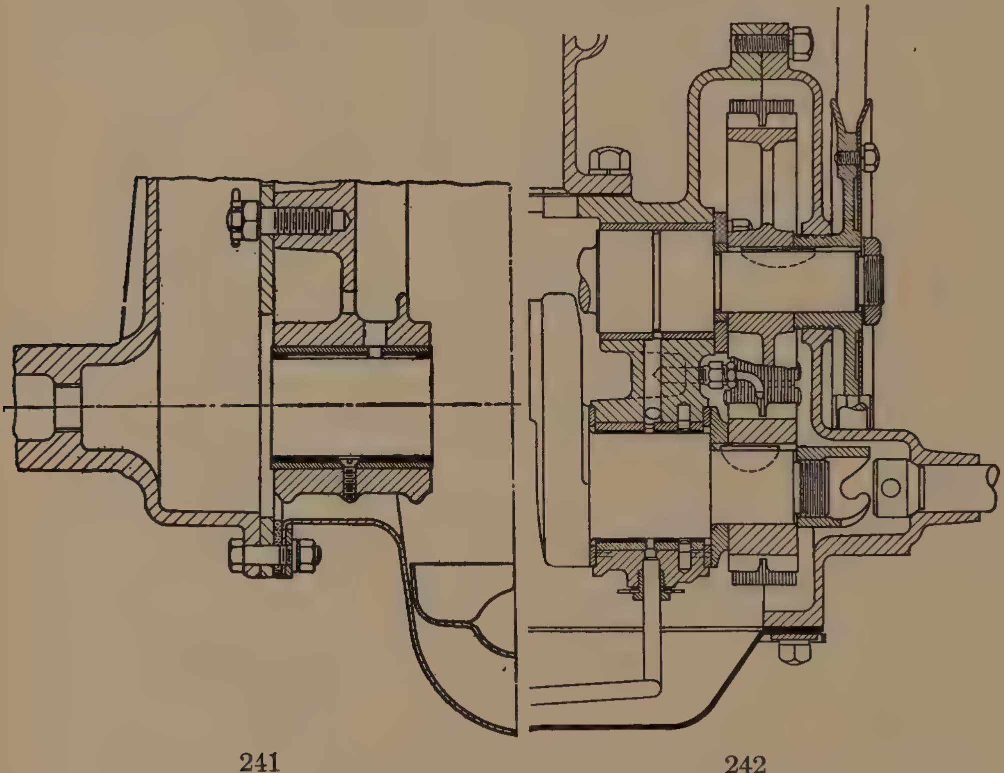


FIG. 240.—FRONT END OF ALL-CAST CRANKCASE.

Front Ends—At the front end of the crankcase there is usually a case or housing for the timing gear or "front end" drive. This may be cast either as an integral part of the crankcase (with the exception of the cover plate) one half of it may be cast integral with the crankcase and the other half formed by the cover plate or removable part, or the whole of the case may be formed by a cast cover which is bolted against either a flat surface formed on the crankcase or against a stamped steel plate, forming the front supporting member of the engine.

The construction of the front end of the crankcase differs according to whether the case is divided on the crankshaft axis or below it, and whether a cast or stamped lower section is used. Fig. 240 shows the front end of a crankcase for a truck engine. The plane of division contains the crank axis, and both sections are castings. At the front the lower section is carried so far forward that the cap of the front main bearing is entirely within it. Then follows a sort of neck which conforms to the contour of the crankshaft pinion, and forward of this there is a flange, the outer, finished surface of which is faced off flush with the timing gear case on the top section of the crankcase. Against this plane surface bolts the cast cover which has a cylindrical extension enclosing the jaws of the starting crank, the forward section of which serves as


241 242

FIGS 241 AND 242.—FRONT ENDS OF CRANKCASES WITH PRESSED STEEL BOTTOMS.

a trunnion support for the engine. The starting crank is supported in a housing that is flange-bolted to the timing gear case cover.

In Fig. 241 is shown the front end of a passenger-car engine crankcase which is parted on the crankshaft axis and has a pressed steel lower section. In this case, also, the crankcase lower section is necked at the forward end, but the neck surrounds the forward main bearing instead of the timing gear pinion. The reason for the difference is that here the crankcase upper section is machined flush with the forward end of the front main bearing, and the timing gears are contained entirely within the detachable cover, which is of housing form. Clamped between this cover and the forward face of the crankcase is a plate steel stamping which serves as the forward supporting member of the engine. Beyond the neck the forward end of the lower section of the crankcase is flanged, the joint with the timing gear-case cover being of the same general form as in the previous example.

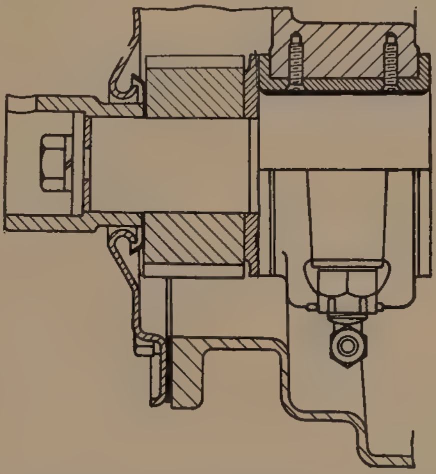


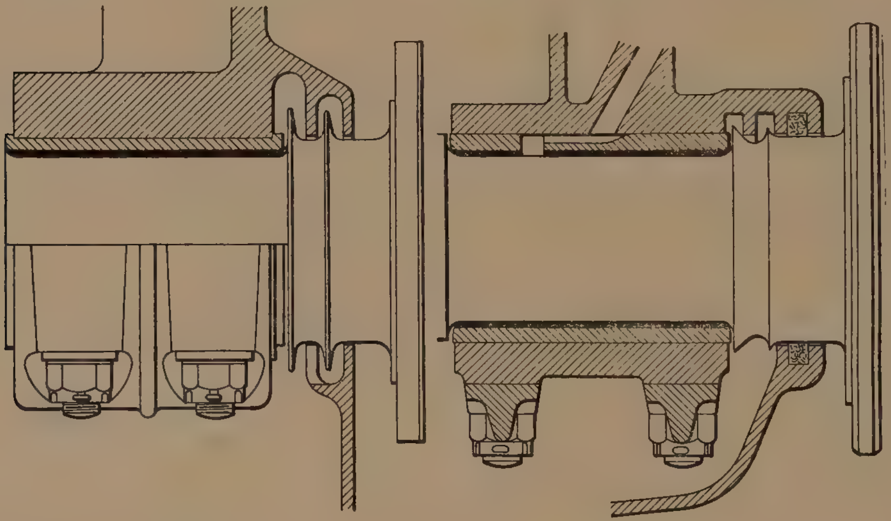
FIG. 243.—FRONT END COMPRISING A PRESSED-STEEL GEAR-CASE COVER.

The front end of a passenger-car engine crankcase divided below the crankshaft axis is illustrated in Fig. 242. In this case the timing gear housing is formed one half in the crankcase casting and one half in the cover. The lugs for the bolts, it will be noted, are on the outside of the housing. Designers of more elaborate cars, for the sake of neatness, make the case sufficiently larger to permit them to place the lugs on the inside. The cover has a machined plane bottom flange, the lower surface of which is flush with the bottom of the upper section of the crankcase, and the pressed steel oil pan has a plane top flange which is bolted against the upper section of the crankcase and the cover plate, with a compressed cork or similar gasket between.

All of the front ends described so far had cast timing gearcase covers, but the one shown in Fig. 243 has a pressed steel cover. The method employed for preventing the escape of oil around the crankshaft where it passes through the cover is of interest, an oil slinger being clamped between the crankshaft pinion and the starting crank clutch jaw.

Rear Ends—Rear ends of two crankcases with cast lower section are shown in Figs. 244 and 245. In both cases the division plane is on the level of the crankshaft axis. The crankshaft in each case has two oil slingers. In the first illustration these alone are depended upon to prevent the oil from getting out of the case along the shaft, while in the second a packing in a grooved section of the bearing is provided as a further safeguard.

Constructions used with pressed steel lower sections are illustrated in Figs. 246 and 247. In the former the plane of division is on the level of the crankshaft axis. A flange is run around the bearing cap at its middle, which extends into a grooved neck at the rear end of the lower section in which is placed a strip of packing material. There are no bolts at the ends of the case to secure the pressed steel pan to the upper section, and in order to ensure an oil-tight joint, the necked portion of the pan is stiffened by a reinforcement as shown. Oil working through the rear bearing is thrown off by a slinger into a groove from which there is a drain to the crankcase, and if any should get by this, it is forced back by an incomplete screw thread fitting snugly in the rear end of the bearing hub and scraping any oil on the bearing forward.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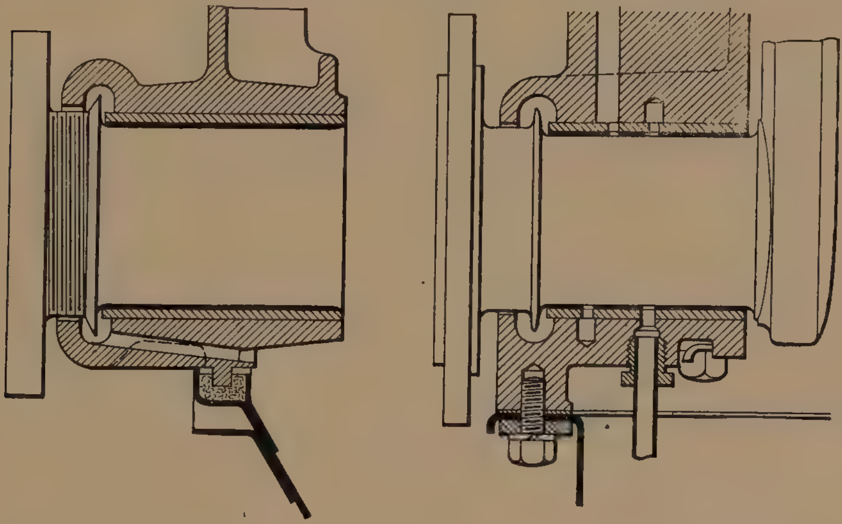
245

FIGS. 244 AND 245.—REAR ENDS OF CRANKCASES WITH CAST LOWER HALVES.

In Fig. 247 the plane of division is below the crankshaft axis and the rear bearing cap is provided with a flange or plate at the rear end which fits tightly in a corresponding slot in the rear wall of the upper section. At the bottom this plate comes flush with the bottom surface of the upper section, and the oil pan, which has a turn-over flange, is bolted against this surface.

Flywheel Housings—The great majority of passenger cars and light trucks are provided with unit power plants, in which the change gear case is bolted to the engine. The engine crankcase is provided with a housing for the flywheel, which may be either cast integral with it or bolted on. In most cases the flywheel housings comprise an upper and a lower section, the plane of division being the same as that of the crankcase. Six sizes of flywheel housing have been standardized by the S. A. E., the items standardized including the outside diameter, inner diameter of chamber, bolt circle diameter, inside diameter of flange and number and size of cap screws. The standard dimensions are given in the following table:



246 247

Figs. 246 and 247.—Rear Ends of Crankcases with Pressed Lower Halves.

The standard specifications further provide that cap screws shall be threaded in accordance with the Free Fit (Class 2) Tolerances of the S. A. E. Standard for Screw Threads, and shall be equally spaced on each side of the vertical center line. The tolerance for the flywheel housing bore shall be plus 0.005 and minus 0.000, and the maximum eccentricity 0.005 in. (indicator reading, 0.010 in.), except for the No. 0 size, which shall be plus 0.010 in. and minus 0.000. The maximum deviation of the face of the flywheel housing flange from its true position, when the housing is rotated on its axis, shall be 0.003 in. (indicator reading, 0.006 in.).

A separate bell housing for an engine is illustrated in Fig. 249. The upper section is secured to the corresponding section of the crankcase by three large bolts and is held accurately in position by two dowel pins. This section is cast with a mounting for the starter. Starter mountings also have been standardized by the S. A. E. and particulars of these standards will be found in the Appendix. If no starter is carried, as on some trucks, a sheet metal plate is bolted over the mounting flange.

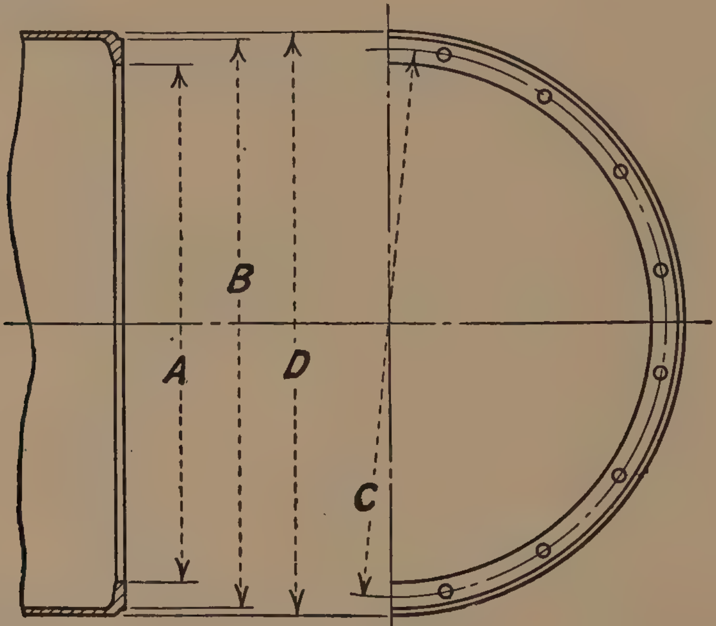


FIG. 248.

Table X—S. A. E. Flywheel Housing Standards

Size No.	A	B	C	D	Cap No.	Screw Size
0	25 ½	28	26 ¾	28 ½	16	½–13
1	20 ⅛	21 ¾	20 ⅞		12	⅜–16
2	17 ⅝	19 ¼	18 ⅜		12	⅜–16
3	16 ⅛	17 ¾	16 ⅞		12	⅜–16
4	14 ¼	15 ⅞	15		12	⅜–16
5	12 ⅜	14	13 ⅛		8	⅜–16

The upper section also carries two integral supporting arms. These arms also have been standardized and a table of standard dimensions will be found in the Appendix. Of course, car and truck manufacturers who make their own engines have no particular incentive to follow this standard, which is used chiefly by manufacturers of engines for the trade.

Typical Aluminum Crankcase—Drawings of an aluminum crankcase for a six-cylinder 3½ by 5 in. passenger-car engine (Pierce-Arrow) are shown in Figs. 250 to 253 inclusive. Both the upper and lower sections are of aluminum alloy, the joint between the two sections being 3⅜ in. below the crankshaft axis. The greater part of both the flywheel housing and the camshaft gear housing is cast with the upper section of the crankcase. There are seven main bearings, and the upper section, therefore, is cast with five bulkheads. Attention should be called to the great depth of the bearing caps, which are of truss form and also cast of aluminum alloy. The rear bearing cap is of rectangular cross-section and forms a closure for a corresponding slot in the upper section, so the joint between the two sections at the rear is in a single plane. At the forward end the lower section is necked and comes flush with the upper section, so there is a plane vertical surface to which the timing gear case cover can be bolted. The lower section of the flywheel housing is a separate casting and has a flanged and bolted joint with the upper section.

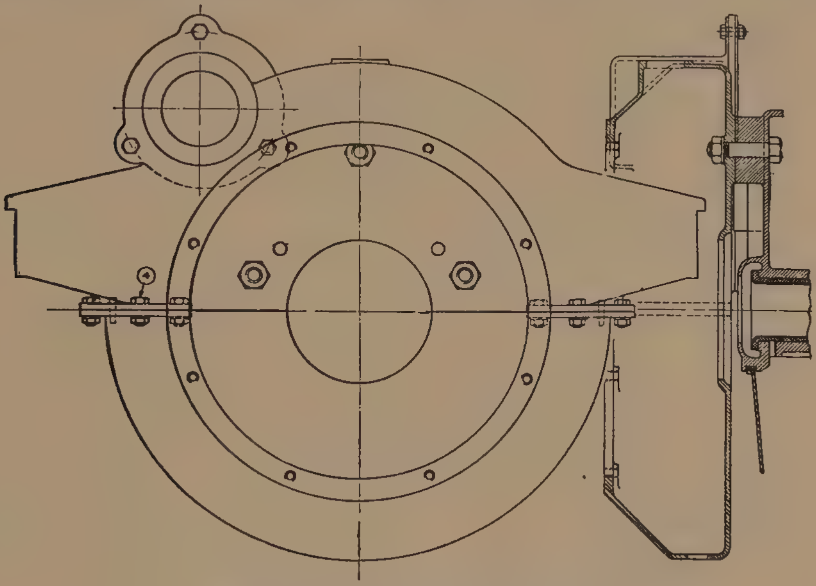


Fig. 249.—Separate Flywheel Housing.

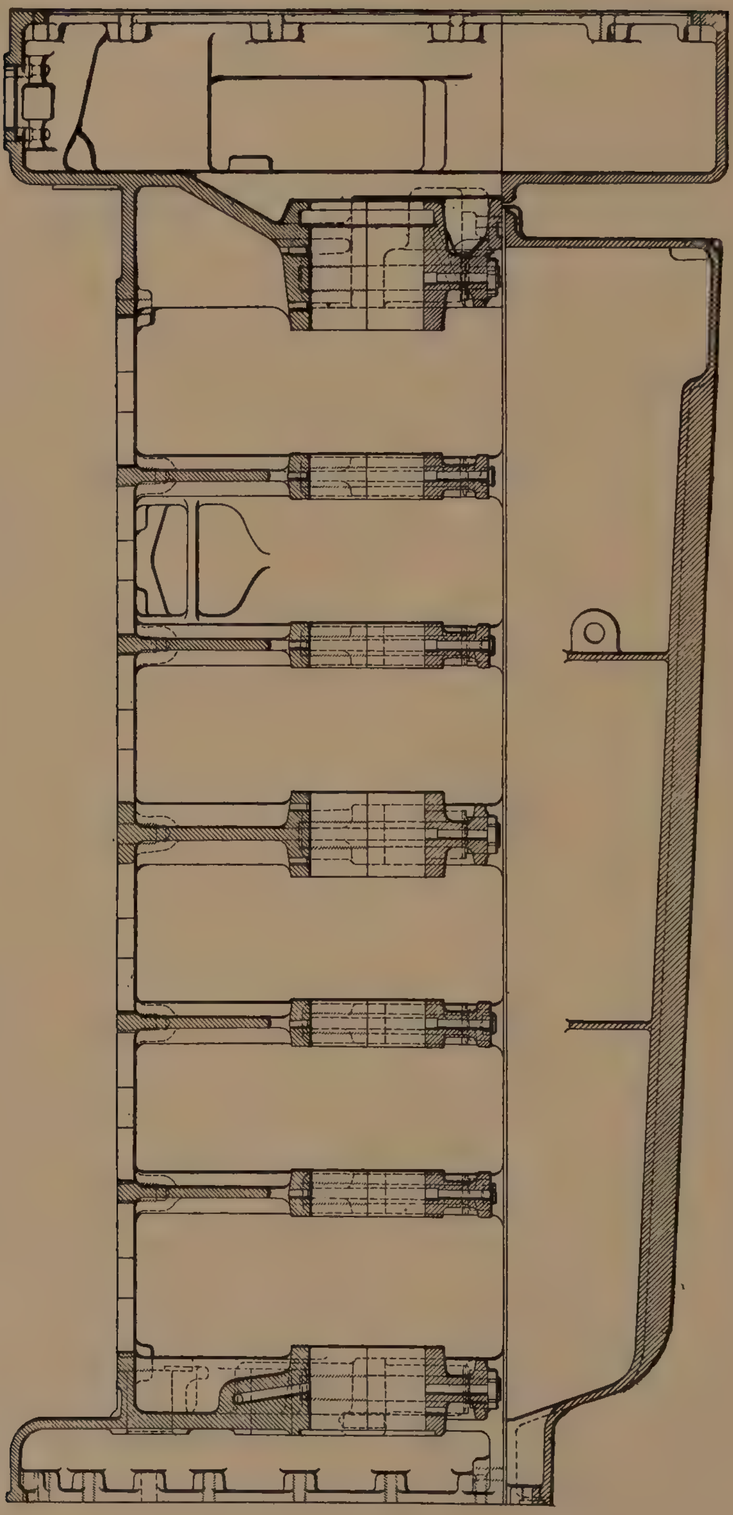


FIG. 250.—LONGITUDINAL SECTION OF SIX-CYLINDER CRANKCASE (PIERCE ARROW).

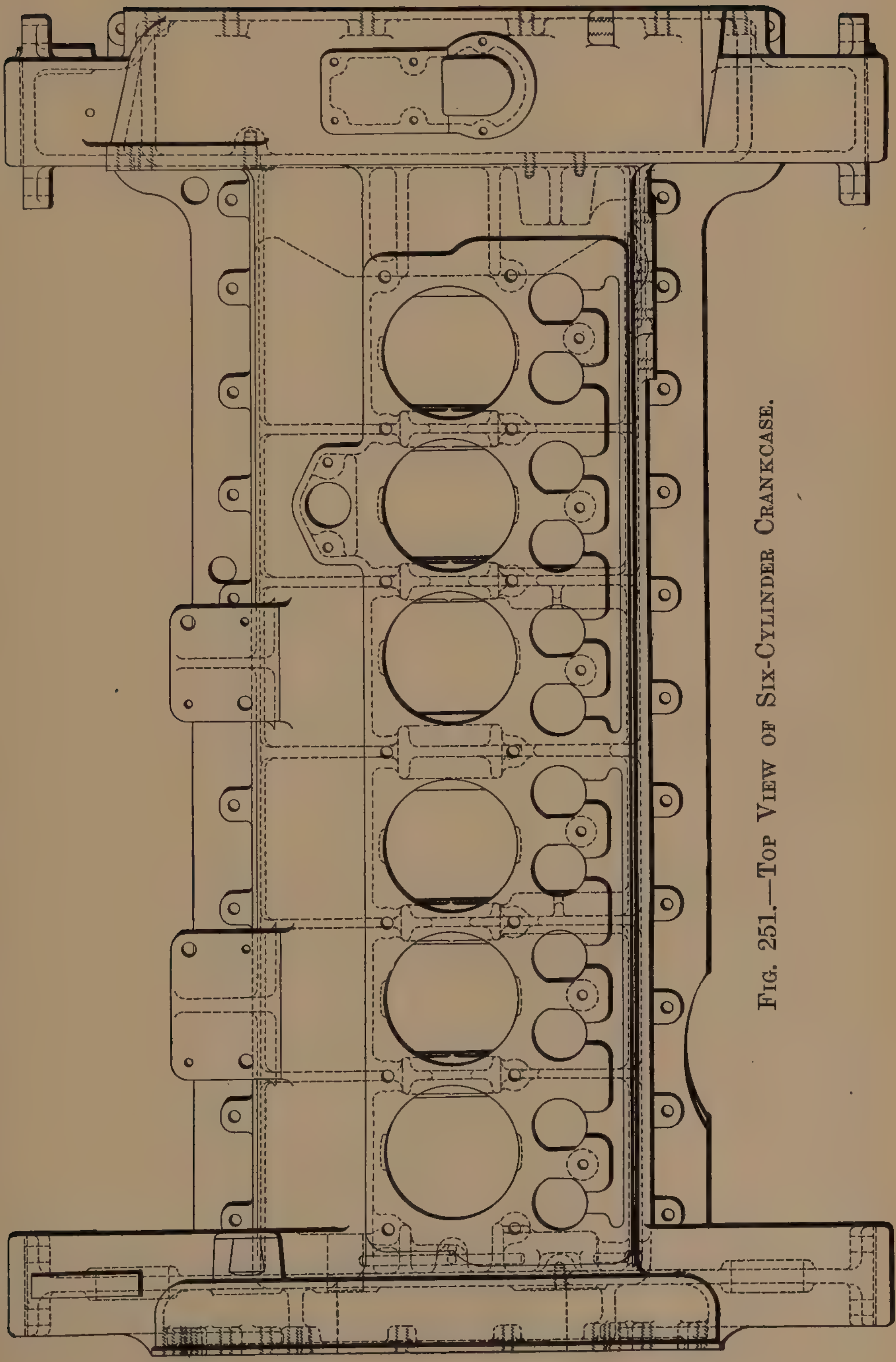


FIG. 251.—TOP VIEW OF SIX-CYLINDER CRANKCASE.

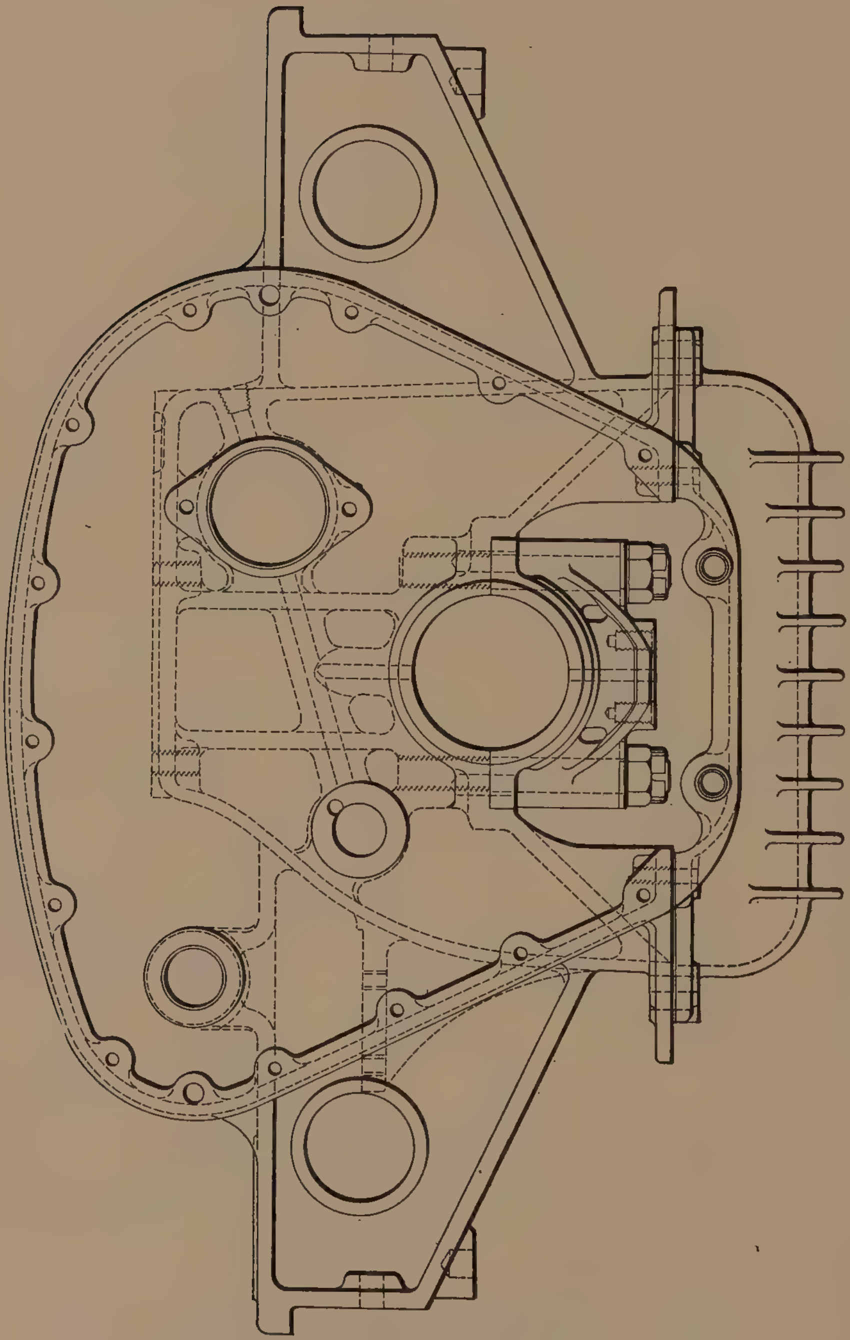


FIG. 252.—FRONT VIEW OF SIX-CYLINDER CRANKCASE.

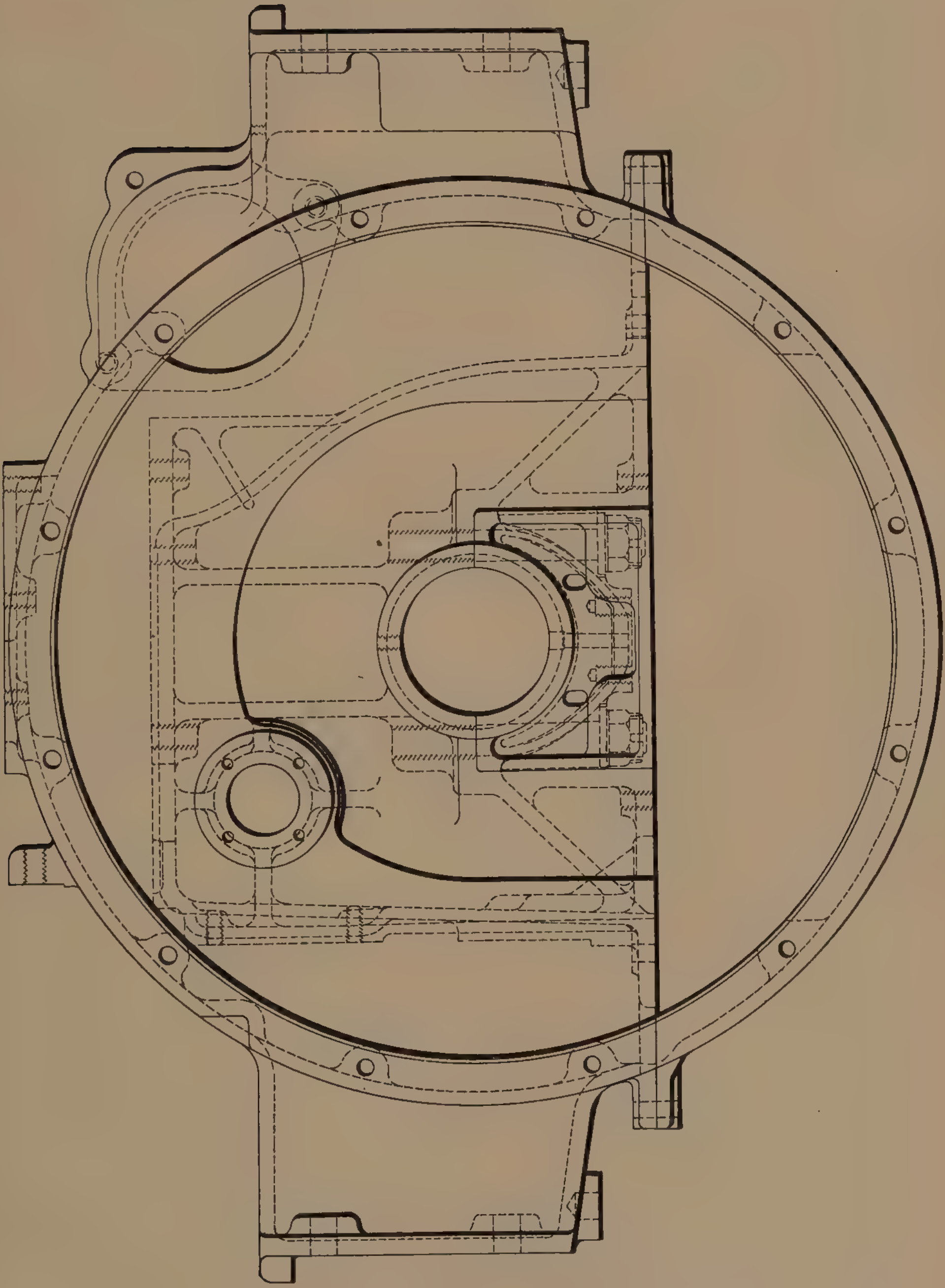


FIG. 253.—REAR VIEW OF SIX-CYLINDER CRANKCASE.

The lower section of the crankcase is cast with oil baffle plates on the inside and oil cooling ribs on the bottom. The two sections are fastened together by means of 5/16-in. bolts at 3-in. centers. Brackets for the pump and generator are cast integral with the upper section on the right side. The rear supporting arms are of box-section, open to the flywheel housing, while the front arms are of I section. A triangular chain front-end drive being used, the front wall of the upper section has cast on it bosses for the main bearing, the camshaft bearing, the accessories driveshaft bearing and the idler stud. The design is remarkably symmetrical. Circular openings are cast in both front crankcase arms, through one of which is passed the hose pipe from the radiator bottom tank to the water pump. To the boss on top of the flywheel housing a control shaft bracket is bolted.

Supporting Arms—Automotive engines have either a four-point or a three-point support. In the former case there are four supporting arms, either cast integral with the upper section of the crankcase or bolted to it. When three-point support is used there are only two supporting arms, generally on the flywheel housing (if one is used), the third point being at the front, where there is a trunnion concentric with the crankshaft, which is carried in a trunnion block on a cross member of the frame. Integrally cast supporting arms should preferably be so located that one of their vertical walls is a direct continuation of a cross wall of the crankcase, as that will insure the greatest strength. The arms frequently are of channel section, with the open side down, but I section arms are also used. Although the height and center-spacing of the supporting pads have been standardized by the S. A. E., these standards are not followed to any extent by vehicle-manufacturers producing their own engines.

Four-point engine support, with the ends of the long, integral supporting arms resting directly on top of the frame side members, is much used in Europe, but is not in favor in this country, for one reason because the long arms make the crankcase a rather unwieldy piece of work. Numerous different supporting methods have been worked out here, with a view to lessening the cost of production, protecting the engine against excessive strains due to frame weave, minimizing the transmission of vibration from the engine to the car and body, and incidentally stiffening the frame.

Three-Point Support—When the three-point support is used the third point is mostly at the forward end of the engine, though the reverse arrangement also has been tried. A very common practice is to provide a substantial cast iron cover for the cam gear housing, turn off a portion of the starting crank hub, and mount it in a trunnion block at the middle of the frame cross member. Such a trunnion with its support was shown in Fig. 240. The general plan of a front trunnion mounting is shown in Fig. 254, though the details there shown are not the most highly developed. The cap of the trunnion block is entirely too light. With heavy truck engines it has been found a good plan to "bush" the trunnion block and

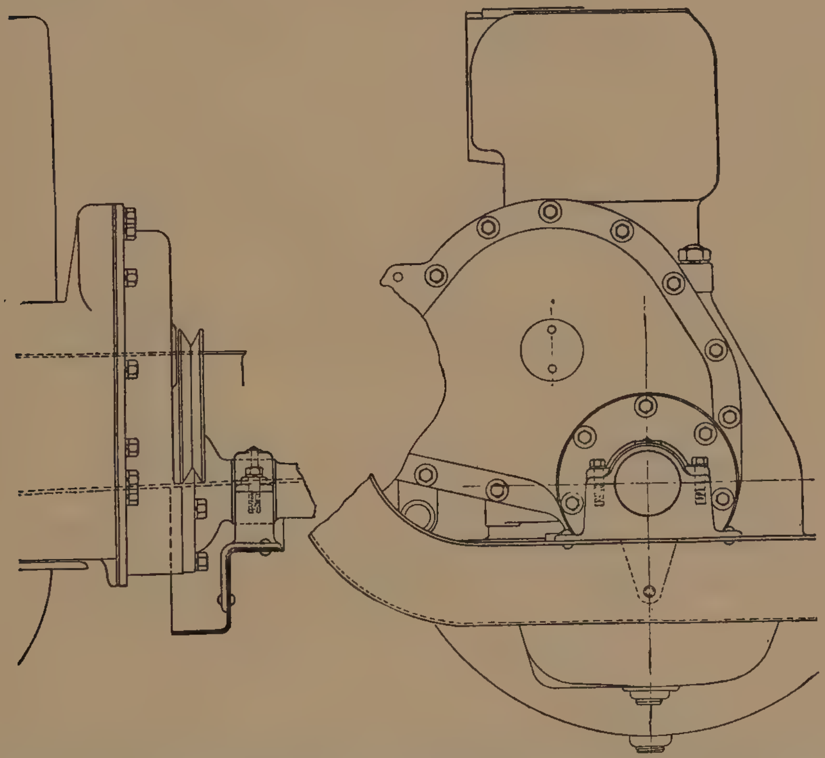


Fig. 254.—Front Trunnion Support of Three-Point Engine Mounting.

provide means of adjustment, and sometimes an oil cup or a grease cup is provided, even though the amount of motion at this point is insignificant. If these precautions are not taken there may be comparatively rapid wear (depending, of course, on the amount of frame weave) and the mounting may become a source of noise.

When thus supported on a trunnion block, the forward end of the engine has both longitudinal and angular freedom in a transverse plane. Some makers have even gone to the extent of making this front support of spherical shape. Dodge

Brothers, for instance, have patented a construction by which a sleeve with a spherical formation at one end is slidingly inserted into the hub formed on the cam gear housing cover, and the spherical portion rests in a corresponding socket in the front engine bearer.

The objection to the supporting methods just described is that they prevent access to the camshaft gears as long as the engine is mounted on the car. This is overcome by having the trunnion bearing on top of the crankcase. As shown in Fig. 255, a yoke may be secured to the crankcase by the bolts holding the cap of the front crankshaft bearing in place. Another plan is to cast the yoke right on the cam gear housing.

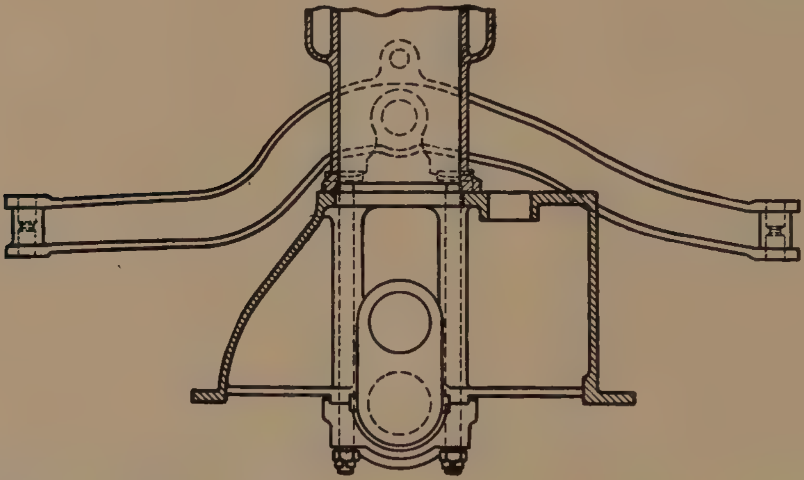
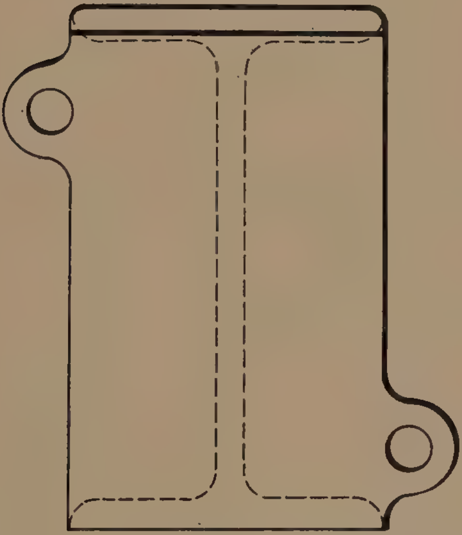


Fig. 255.—Front Trunnion Support which does not Interfere with Access to Camshaft Gears.

The fan bracket may then be secured into a lug formed in the engine bearer or bolted to it.

At the rear the engine is supported by arms which may be either cast integral with or bolted to the crankcase or flywheel housing. One method of support consists in having these arms extend right up to the frame side bars. A pressed steel or cast filler is riveted into the channel of the frame member where the crankcase arm is bolted to it. The end pad of the supporting arm is generally of substantially rectangular form, of about the same depth as the frame member and somewhat lesser width, and on each side of this pad there is a lug, through which and the frame member (and possibly also the filler) a retaining bolt may be passed, as shown in Fig. 256. The reason for the lugs is that they bring the nuts on the retaining bolts into more accessible positions.

A popular method, however, consists in riveting a bracket to each frame member and bolting the crankcase arms down to these brackets, as illustrated in Fig. 257. This permits of lowering the engine right down onto the frame, and it also facilitates its removal. Similar supporting brackets may be used with four-point support, but at least one of the supports is then made slightly yieldable, so the arms will not be over strained by excessive weaving of the frame. This is accomplished by placing a stiff coiled spring under the nut of one of the retaining bolts, as shown in Fig. 257. Recently the use of rubber blocks and rubber bushings and washers for mounting engines has gained a certain vogue, and two applications of this principle are illustrated in Fig. 258.



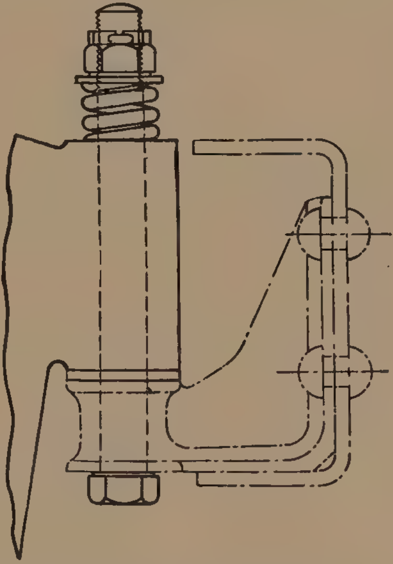


Fig. 256 (left).—Crankshaft Arm Pad.
Fig. 257 (right).—Spring-Cushioned Bracket Mounting.

Drop Forged Engine Bearers—Another method of doing away with the crankcase arms—which are a hindrance while the crankcase passes through the shop and sometimes break in service—consists in using forged engine bearers. There are essentially two methods of supporting the engine by means of forged steel bearers. With the first, both the front and the rear bearer are rigidly bolted to the engine, generally by means of "through" bolts which also hold on the bearing caps of the two end bearings. By the other method the front bearer has a swivel joint with a yoke either cast onto the top of the camshaft gear housing or held in place by the bolts securing the cap of the crankshaft forward bearing. A design of the first-mentioned class, for a truck engine, is illustrated in Fig. 259. The bearers, which are of I beam section, are drop-forged from 0.30–0.40 per cent carbon steel, which is heat-treated to show 70,000 lbs. elastic limit. The rear bearer is rigidly secured to the crankcase by means of four bolts, the two center bolts running through to the under side of the rear bearing cap, with a steel plate under the cap. In addition, two outer bolts are used to take care of the torque reaction and give greater rigidity.

The front support is bolted to the engine crankcase by the two bolts holding the forward main bearing cap. These bolts are comparatively close together and give a modified three-point support to the engine. This form of engine support has the further advantage of considerable elasticity, and although the crankshaft is supported in seven bearings and the clearance of these bearings is only 0.0015 to 0.002 inch, there is no binding of the crankshaft when one of the front wheels is raised 8 inches from the ground.

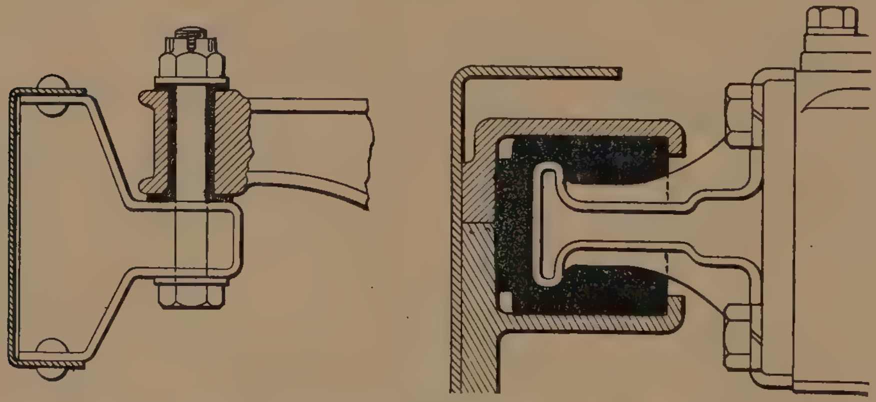


Fig. 258.—Engine Mounting on Rubber Washers or Blocks (Chrysler).

Metal filling blocks are fitted in the frame side channels and two bolts in each end of each engine bearer secure the engine to the frame. The top flange of the engine bearer overhangs the filler block and therefore relieves the bolts of the weight of the engine. The steel engine bearers make very strong frame cross members and permit of the use of a light front cross member which carries the radiator. The use of a forged engine bearer together with a trunnion mounting was illustrated in Fig. 255.

Pressed Steel Engine Bearers—Recently punched or pressed steel engine bearers have come into extensive use on

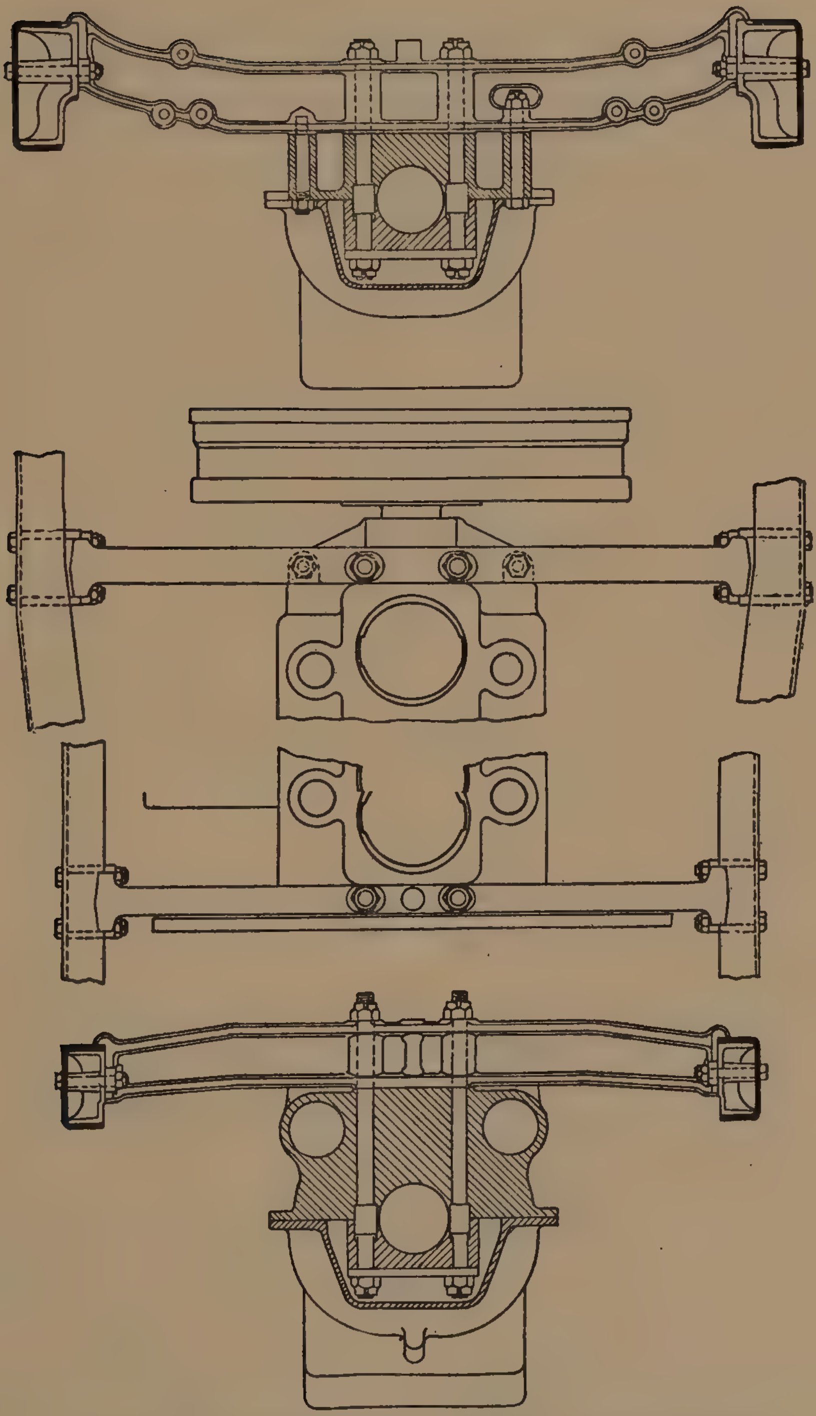


FIG. 259.—ENGINE SUPPORTED BY DROP FORGED BEARERS.

small, low-priced cars. These punchings are clamped between the crankcase and the timing gear cover plate in front, and between the flywheel housing and the clutch housing in the rear. Various methods of fastening these plates to the frame side members are in use, that illustrated in Fig. 260 having been used by Chevrolet. The ends of the supporting plates or beams rest on a supporting ledge formed on a filler block in the frame channel. The filler block is riveted to the frame and the supporting beam is held in place by means of a U-bolt passing through a slot punched in it and through holes drilled in the frame and the filler block. A more common plan, however, is to use brackets which are riveted to the frame and which have the plates bolted to them. If punched or pressed steel bearers are used at both ends of the engine, the rear one usually is flanged for increased stiffness in the fore and aft direction.

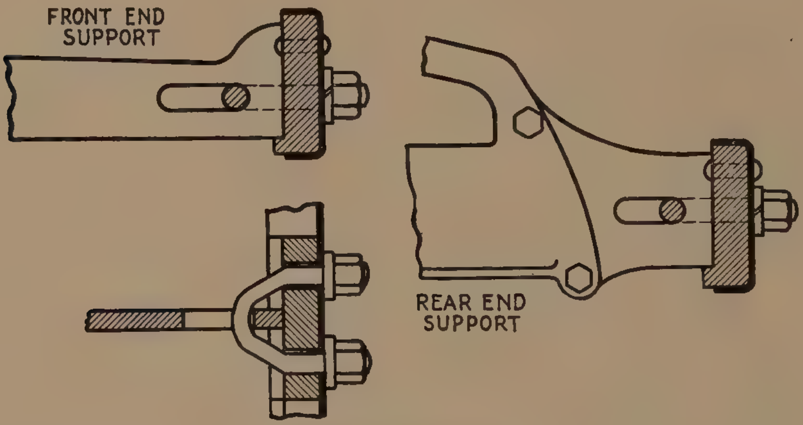


Fig. 260.—Front and Rear Supports by Steel-Plate Bearers.

Breathers and Oil Fillers—In the crankcases of four-cylinder engines there is a regular pumping or breathing action, owing to the alternate increase and decrease in volume resulting from the difference in the momentary speed of the ascending and descending pistons. To permit of this breathing action the crankcase must be vented. In the conventional six-cylinder engine, the vertical eight and other similar types, this breathing action is absent, but the crankcase must still be vented, because of the leakage of gases of combustion past the pistons. If the crankcase were not vented these would build up a high gas pressure in it, which would tend to force the oil out through the bearings, joints of the case, etc. Hence, the crankcases of all types of engine are provided with so-called breathers, which, in the case of multiple cylinder engines are mere vents, however.

The pumping or breathing action is strongest in a single cylinder engine as used on motorcycles, in which the crank chamber volume increases and decreases by the full amount of the piston displacement during every revolution of the crankshaft. The crankcases of such engines are provided with a valve, either automatic or mechanically operated, which opens when the crankcase volume decreases and closes while it increases. A slight degree of vacuum is therefore maintained in the crankcase, and this tends to keep the oil in.

As a rule the breather or vent serves also as an oil filler. On four-cylinder and other types of crankcase in which there is a pronounced breathing action, the breather must be efficient as an oil separator, for the air in the crankcase always carries in suspension a considerable amount of oil, and if this were not trapped, much oil would be lost and the space under the engine bonnet would become quite dirty.

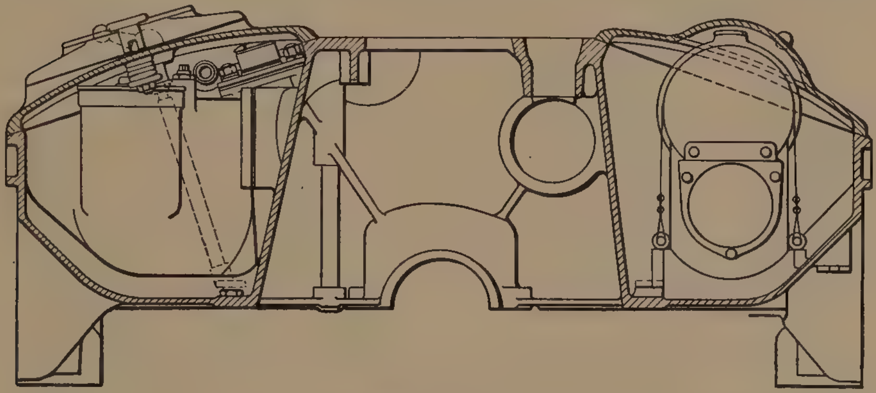


FIG. 261.—FIAT CRANKCASE UPPER HALF WITH CHESTS FOR ACCESSORIES.

Continuous Webs—With aluminum crankcases, and particularly with those having four integral supporting arms resting on the frame side bars, the flange at the bottom of the upper section is sometimes extended to the frame side member, thus closing the engine space off from below and making a sheet metal mudpan under the engine unnecessary. This so-called continuous web construction has been popular in Europe for a long time. The webs on both sides of the crankcase extend from the front to the rear supporting arm and form a base for such accessories as the water pump, magneto, etc. In some cases large chests are formed integral with the crankcase in which the engine accessories are placed, access to them being had through covered openings. In Europe the practice of selling chassis to the ultimate consumer still prevails to a large extent, and great efforts are therefore made to obtain a neat chassis, as devoid as possible of any semblance of complication.

Manufacture of Crankcases—In quantity manufacture, aluminum crankcase castings are generally produced in the foundry with molding machines and from metal patterns, so they come very close to size. The castings are closely inspected, and if any defects show up these generally can be remedied by means of the acetylene torch. Salvaging defective castings in this way is a recognized procedure. In some cases, after inspection the castings are sprayed with a japan on the inside, so as to close up all pores and make the cases oil-tight.

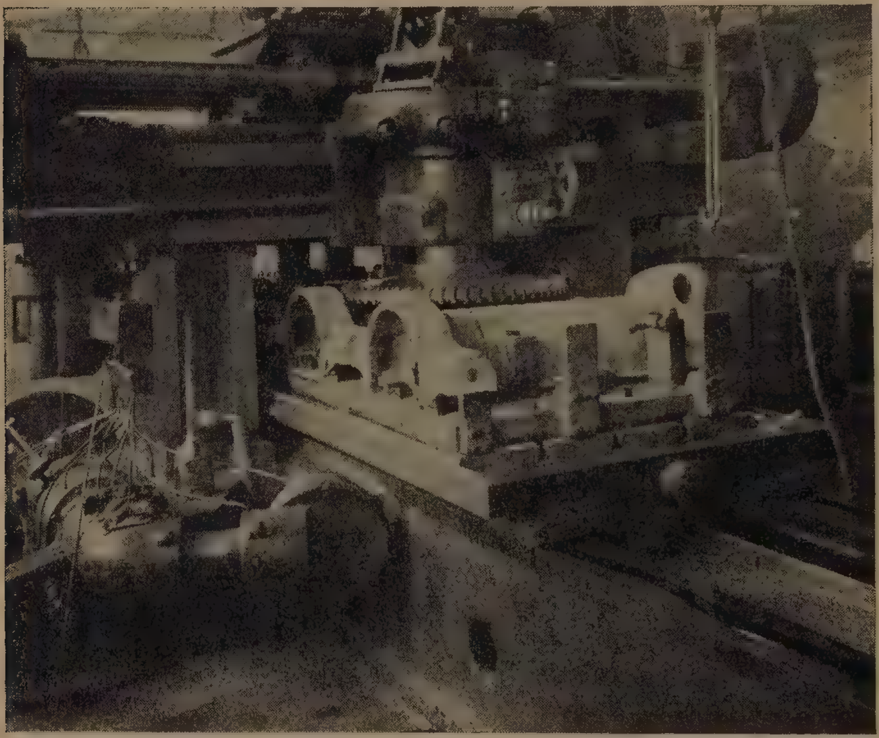


FIG. 262.—SURFACE-MILLING TOP OF ALUMINUM CRANKCASE.

The first machining operation generally consists in surface-milling the bottom flange, together with the faces of the bearings and the recesses for the caps. A number of cases are strapped down to the table of a large milling machine carrying six or more cutters, and both roughing and finishing cuts are taken at the same pass, the cutters on the forward or entering side performing the roughing cuts and the rear cutters the finishing cuts.

Next the cases are loaded into another continuous milling machine where the top surface is rough-milled and both end faces are straddle-milled—that is, roughed. Fig. 262 shows a crankcase undergoing this operation in the Hupmobile plant, the cutter for the top surface having 65 teeth. Sometimes two cutters, each of about half the diameter as this one, are used for the top surface. The main bearings usually are straddle-milled for length and correct spacing, and at the same time the oil slinger groove at the rear end is finished.

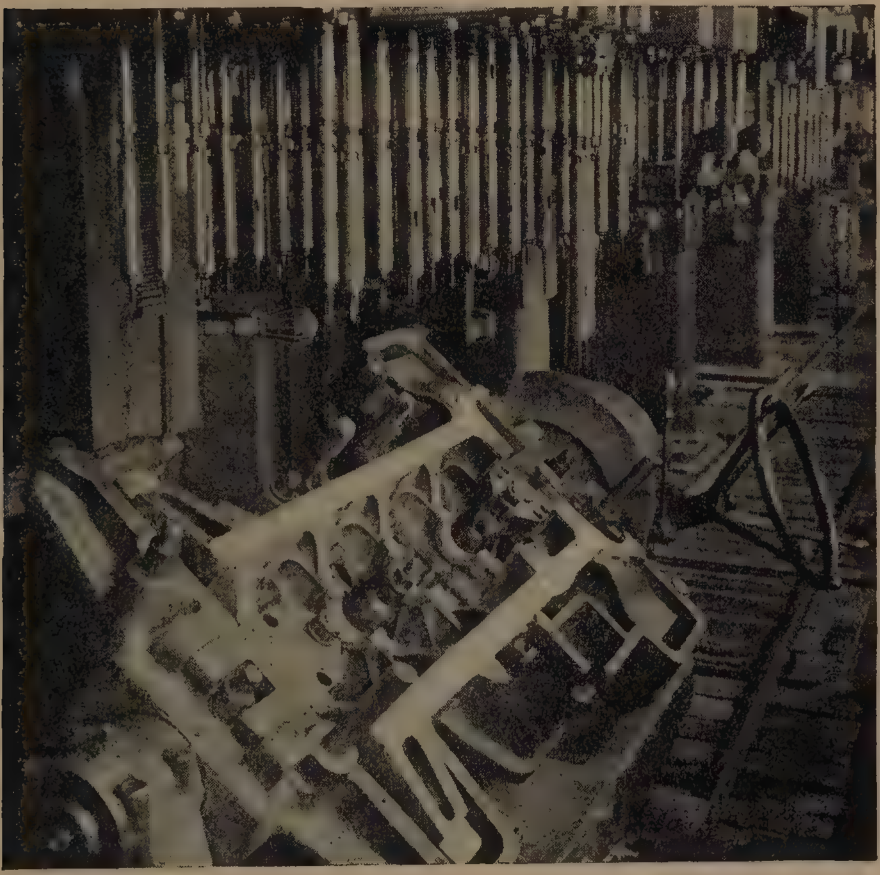


Fig. 263.—Railroad Gang-Drill for Crankcase Drilling in Hupmobile Plant.

Next follow a number of drilling operations. All of the holes for the bearing caps are drilled in one operation in a multiple spindle machine, and a little later on they are all tapped in a similar machine. For the drilling and tapping of the holes in the outer surfaces of the case two methods are available—the multi-way drill with box-jig in which holes are drilled in both ends, top and bottom at the same time, after which the holes in all except the bottom flange are tapped simultaneously in a three-way automatic tapper; or the drilling may be done in a railroad gang drill, with the case held in a tumbling jig carried on a fixture provided with rollers for moving it along the track from one multiple spindle drill to the next and then to the tappers. The principles involved are the same as those described in connection with the manufacture of cylinder blocks. The crankcase drilling and tapping line at the Hupmobile plant is shown in Fig. 263.

Next follows the boring of the bearings, and here, too, different types of equipment are used. Previous to this operation, of course, the caps must have been fitted to the main bearings. Generally all of the bearings are bored out in one operation, this including the main and camshaft bearings, the accessories drive shaft bearing or bearings, and, possibly, a hole for the stud of an idler sprocket or gear. The rough-boring is followed by a finish-boring. Fig. 264 illustrates this operation on the Hupmobile crankcase. The methods of fastening down the crankcase with heavy hinged arms, and the type of cutter used are clearly shown. Sometimes the finish-boring is followed by a finish-reaming of the main bearings, in order to obtain the best possible contact between the case and bushing, so as to ensure free heat flow.

Next the bronze-back, babbitt-lined bushings are pressed in place, care being taken not to mar the babbitt surface, and the dowel pins or screws are inserted. A light cut is then taken from the babbitt by means of a fly-cutter, while at the same time the other bearings are finish-bored.

In some crankcases the oil leads are cast in. Continental Motors Corp., for instance, prepare an oil-distributor assembly of steel tubes, brazed together, which is set into the mold and cast in place. Where this practice is followed the oil lines must be tested for oil-tightness. Otherwise oil passages are drilled, as from the main to the camshaft bearings. Finally the crankcases are washed and all bearings are line-reamed.

Special care must be taken with aluminum crankcases to get all important surfaces accurate as, owing to the release of internal strains upon the removal of any metal, they are apt to change in shape slightly during each machining operation. The Continental Motors Corp., therefore, take a very light finish cut off the top surface after all other machining operations on the case proper have been completed, and machine the flywheel housing flange only after the engine is assembled, by means of a cutter secured to the flywheel and driven through the crankshaft.

Bearing Bushings—Practically all of the bushings for main bearings are of what is known as the bronze-back, babbitt-lined type. As explained in the Chapter on Connecting Rods, the connecting rod big end is now rarely provided with a separate bushing, the babbitt being cast directly into the bore of the rod, with a tin bond, and as the "small end" or piston pin bearing bushing is a plain bronze bushing, only the main bearing bushings need be considered here.

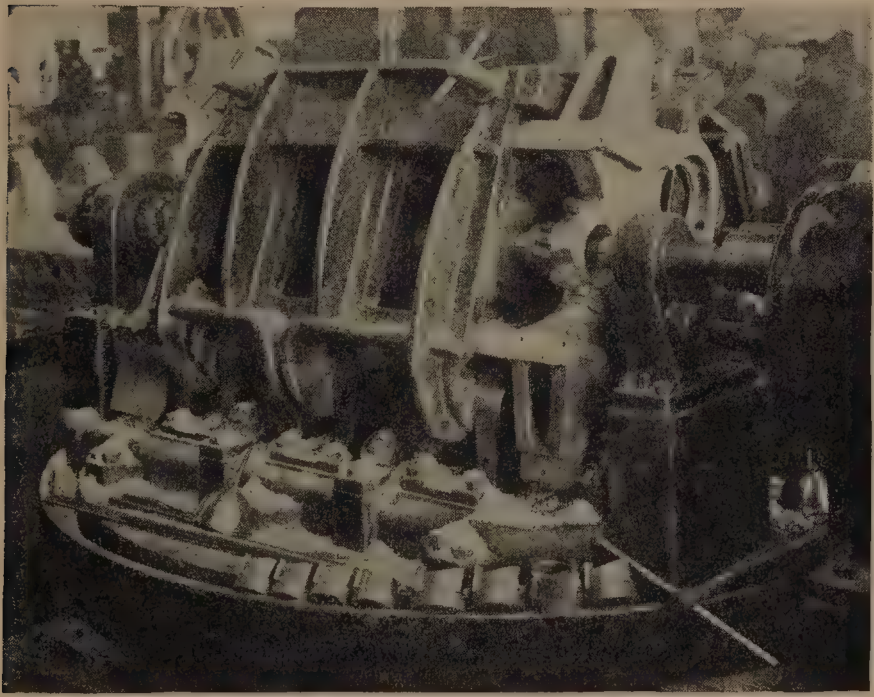


Fig. 264.—Boring Out Crankcase Bearings.

There have been great improvements in the methods of production of these bushings or bearings in recent years, and it is now possible to make them to such close tolerances that they are interchangeable. That is, if a bearing bushing has worn so much that the clearance between it and the journal has become excessive, it can be removed from the engine and replaced by a new one without any laborious fitting. Formerly it was necessary in such a case to produce the proper fit by means of scraping tools and Prussian blue. By covering the surface of the bearing with this latter material, putting the crankshaft in place and rotating it, the blue would be dragged off the high spots, and these would then be removed by the scraping tool. It is not to be understood that these interchangeable bearings are in universal use, but they have been developed to a commercial state and are being fitted by prominent manufacturers in regular production.

Bearing Materials—For the bronze shell of the bearings the S. A. E. No. 66 alloy is used, which is composed of 83–86 per cent copper, 4.50–6.00 per cent tin, 8–10 per cent lead, not over 2.00 per cent zinc and not over 0.25 per cent impurities. This shell, of course, is used to give the bearing additional strength, while the white metal or babbitt lining is used on account of its superior bearing qualities. Five grades of babbitt have been standardized by the S. A. E., but only high tin content babbits are suitable for the linings of main bearings, and these are covered by the following two specifications:

Table XI—Compositions of S. A. E. High Tin Content Bearing Alloys

No.	10		11	
	Cast Products	Ingots	Cast Products	Ingots
Tin (min.)	90.00	90.75	86.00	87.25
Copper	4–5	4.25–4.75	5.00–6.50	5.50–6.00
Antimony	4–5	4.25–4.75	6.00–7.50	6.50–7.00
Lead (max.)	0.35	0.35	0.35	0.35
Iron (max.)	0.08	0.08	0.08	0.08
Arsenic (max.)	0.10	0.10	0.10	0.10
Bismuth (max.)	0.08	0.08	0.08	0.08
Zinc and alum.	none	none	none	none

When finished bronze-back bearings are purchased, a maximum of 0.6 per cent of lead is permissible in scraped samples, provided a lead-tin solder has been used in bonding the bronze and babbitt. The No. 10 metal is very fluid and therefore is particularly recommended for use where the lining has to be very thin, as in aircraft engines. The No. 11 is a harder babbitt and specially adapted for lining connecting rod heads and shaft bearings subjected to heavy loads. It has only a very slight tendency to wipe.

Form of Bushings—Formerly all bushings or bearings were made with outside flanges at both ends. At present only the bearing on which the end thrust is taken is provided with flanges, and sometimes not even that, as separate thrust collars may be set into turned recesses and secured to opposite ends of the front bearing in the crankcase. All other bearings are plain cylindrical shells. They are made of only such length that the fillet on the crank journals is entirely outside them, which simplifies their production and does not reduce the effective bearing area. The thickness of the bushing wall is generally made about one-tenth the journal diameter, and the width and height of the flanges (if any), the same. The babbitt lining is made as thin as possible, because the greater the proportion of bronze, the stronger the bearing will be, and the less its cost for material. For a journal diameter of 2 inches, a lining 3/64 in. thick can be used, so that if the oil grooves are made 1/32 in. deep they do not cut quite through the lining.

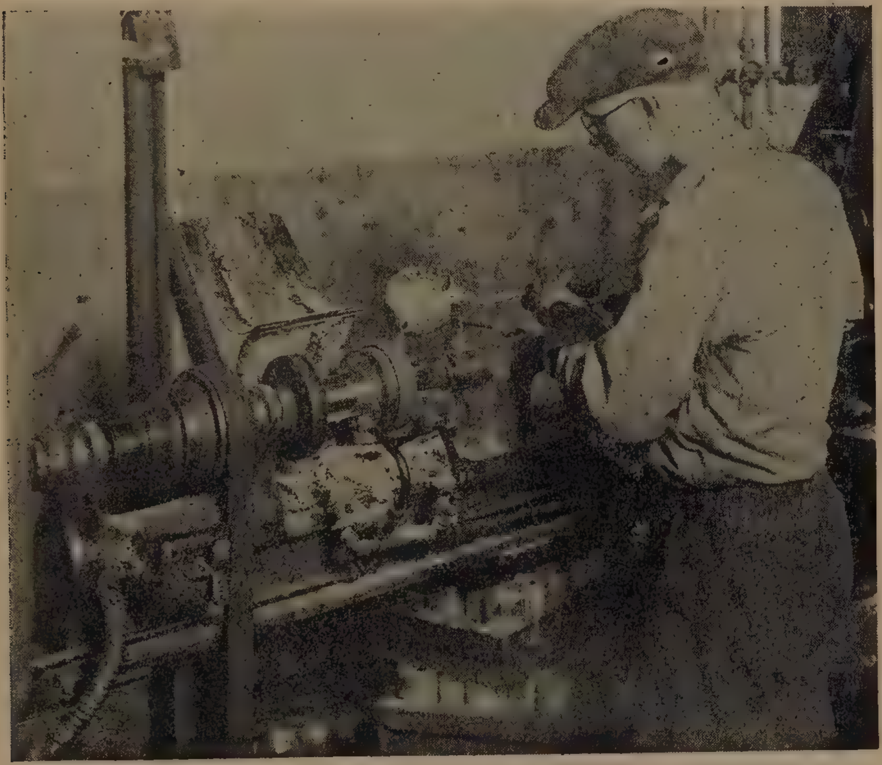


Fig. 265.—Babbitting Bearing Shells by the Centrifugal Method.

Manufacture of Bearing Bushings—Following is an outline of the methods of bearing production employed by the Olds Motor Works, which are characteristic of modern practice in this particular line.

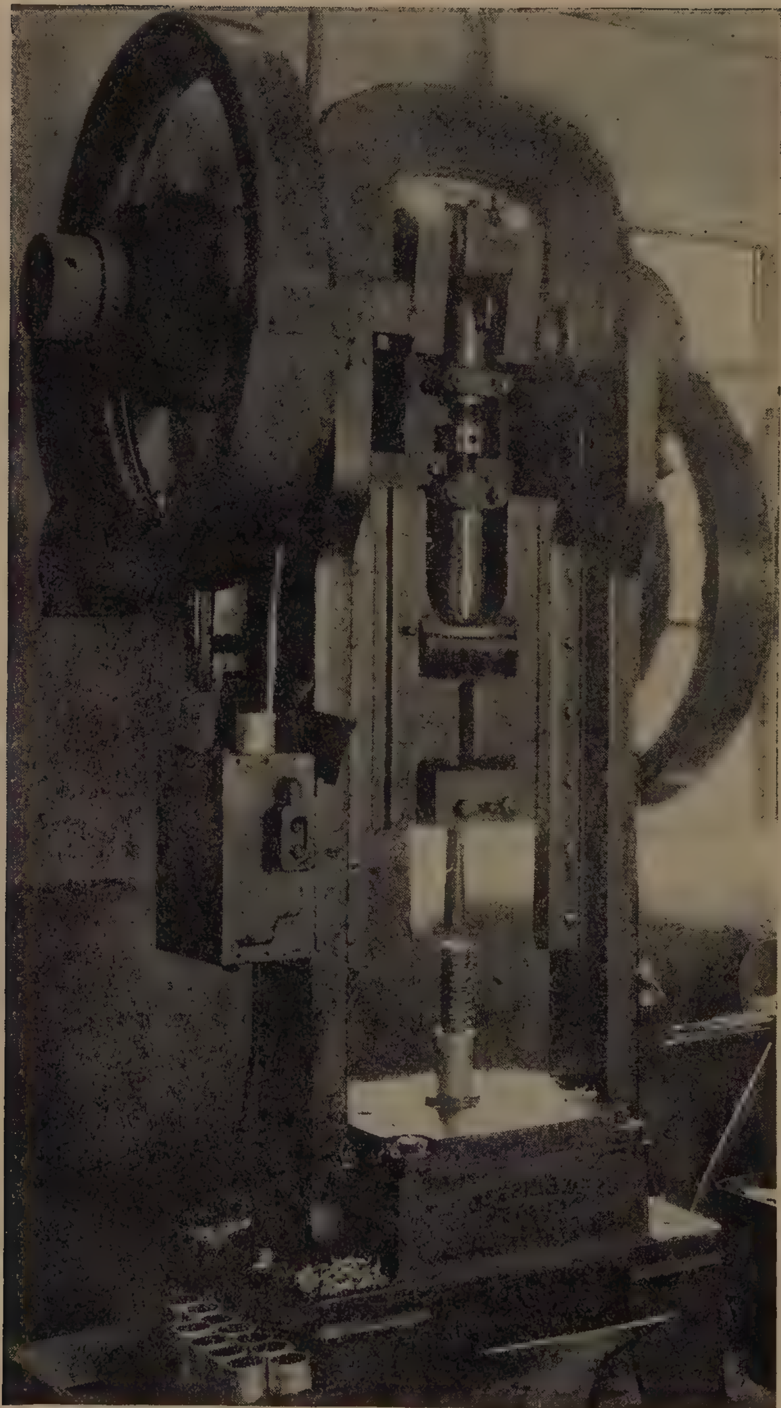


FIG. 266.—BROACHING AND BURNISHING BABBITT-LINED BEARINGS.

The cast bronze shells are first rough-bored and reamed in a No. 4 Warner & Swasey hand screw machine, equipped with air chuck. The reamed bushings are next placed in a Reed-Prentice lathe with an expanding mandrel operated by an air cylinder and a special back arm attachment for back-facing and chamfering. One tool post on the carriage holds the turning tool, while another carries a chamfering tool that comes into action just as the carriage hits the stop at the end of the cut. Meanwhile the back arm, carrying two facing tools, has swung down and faced the ends of the blank. Another tool, hinged to the headstock, chamfers the back end, being swung into position by the carriage just before reaching the end of its travel.

The blanks being now completely machined, they are ready for the babbitting operation. They are first given an acid bath and are then tinned by dipping into a bath of molten tin kept at a temperature of 625° F. in a gas-fired furnace. Both the tin bath and the babbitt, which latter is heated in another gas-fired furnace, are kept at constant temperatures by automatic pyrometer control. The shell is tinned all over, and in the process it attains a temperature substantially the same as that of the tin bath. It is then transferred to a specially equipped speed lathe (Fig. 265) and held between rings carried by the headstock and tailstock, the ring on the latter being mounted on a ball bearing so as to revolve freely. These rings, together with the shell, form a mold into which the babbitt, which is maintained at a temperature of 825° F. is poured in the proper quantity, with a ladle, through an opening in the tailstock. After a short interval to allow the babbitt to level off and dross to rise to the surface, the lathe is started and the shell revolved at 900 r.p.m. At the same time a jet of air is turned onto the outside of the shell and also directed into its interior through the opening in the tailstock. Enough babbitt is poured in to produce a lining 3/16 in. thick, of which three-fourths is subsequently removed by machining. Owing to the high pressure on the babbitt near the bond, due to centrifugal force (which is calculated to be about 52 lbs. per sq. in.) and the rapid cooling, a very dense lining is produced.

Next the babbitted bearing is bored, roughing and finishing cuts being taken successively in a turret lathe. A short broach with 12 cutting and three burnishing lands is then forced through the shell in a Bliss press (Fig. 266). This is so equipped that as the ram forces the broach down through the bed, it is caught by a spring table and automatically returned to a position where it can be readily grasped by the

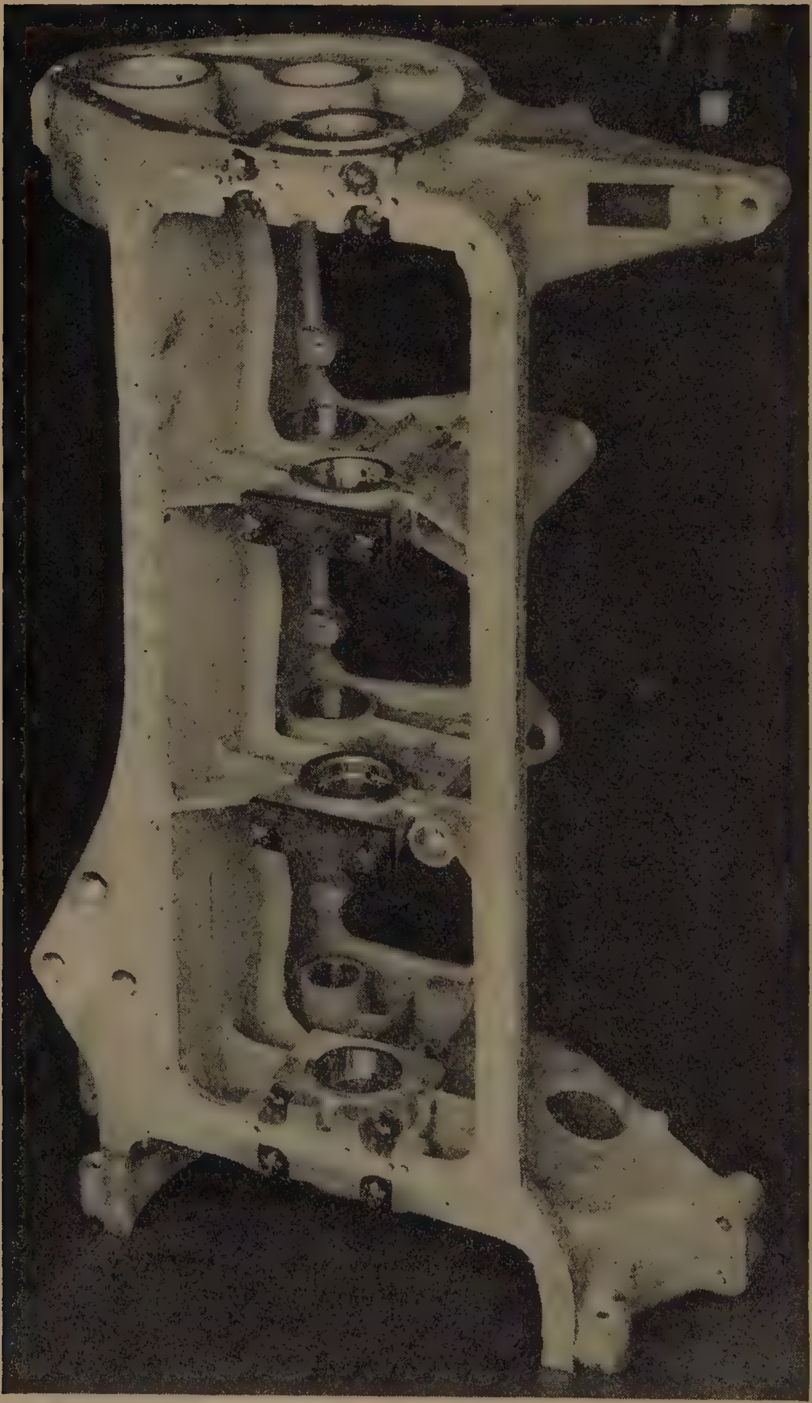


FIG. 267.—COMPLETED CRANKCASE WITH BEARINGS IN PLACE (HUDSON).

operator. The broaching and burnishing operation completes the bearing surface.

Up to this point the bushings for the rear and center bearings have been in a single piece, and they are now cut apart by means of a circular cutter. Next the outside of the shell is finish-turned, this operation, performed in a Reed-Prentice lathe, being practically identical with the original rough-turning operation. The shells are then ground on the outside by means of a cylindrical grinder with a wheel of full width, with a straight-in feed.

Splitting of the shells is effected in a hand mill with a 0.038-in. slitting saw. They are clamped in a fixture mounted on the table, and stops are provided on the column so that the table can be quickly moved from one level to another. In one position the shell is split on top and in the other at the bottom.

This operation leaves the shells slightly above finished size. They are therefore placed in a toggle type fixture by which they are pressed into the proper cylindrical shape circumferentially, and while thus held they are struck a blow endwise with a properly shaped shoe on the ram of a Bliss press. In this way the length is reduced by about 0.006 in. and the contour is fixed. After this the inner and outer edges are lightly chamfered in a hand mill. Owing to the fact that the forming operation affects the contour only, some excess metal is left at the joint surface, and this is removed by means of face broaches in a punch press. A slide guided by ways on the fixture and actuated by the ram, carries face broaches with a wedge adjustment. Above the broaches there are two adjustable blades which "break" the outer corners, following the broaching cut. During this operation the shell is held in a toggle clamp. This completes the bearing, except for the cutting of the oil grooves and dowel holes. The completed bearings are inspected for contour, straightness, wall thickness and effectiveness of bond, the latter being a test by sound. A clear tone indicates a good bond; a flat tone, a defect in the bond.

CHAPTER XV

Engine Lubrication

Contacting metal surfaces of high speed machinery between which there is relative motion must always be well lubricated, in order that the friction accompanying the motion may be slight and that there may be no scoring or cutting when the pressure of contact is high. In gasoline engines oil must be fed to all of the so-called bearing surfaces continuously while the engine is running, in adequate quantity, yet it must not be supplied in excess to the cylinder walls, as the suction due to the partial vacuum in the cylinders during the inlet stroke tends to draw it up into the combustion chambers, especially when running on low throttle, and if too much is fed the exhaust will be smoky with oil vapor, and carbon deposits will form on the combustion chamber walls. For engine lubrication, mineral or petroleum oil is used almost exclusively, though some use has been made also of castor oil, which has the advantage that it retains its viscosity better at high temperatures and thus is particularly suited for use in air-cooled aircraft engines.

Oil may be fed to a bearing in two ways. It may be either delivered into an oil hole or oil pocket located above the bearing and allowed to drain from there to the bearing surface, or it may be fed directly to the bearing surface under pressure. The first is the essential principle of the splash system of lubrication; the second that of the pressure system. There are several modifications of both systems, and they may also be combined in various ways.. The more popular systems will be briefly described.

Splash Lubrication—The simple splash system was much used on earlier models of car. The crankcase contained a supply of oil into which the connecting rod head dipped at each revolution of the crank, thereby splashing the oil over all the interior parts. As the oil worked out through the bearings or past the pistons, the level in the crank chamber fell and the splash became less profuse. When the operator considered that the oil level had become too low, he would replenish the supply by either pouring oil into the crank chamber through a filling hole or transferring it from a supply tank to the crank chamber by means of a hand pump provided for the purpose.

The disadvantages of this system are apparent. The oil level varied constantly and so, naturally, did the rate of supply of lubricant to the bearing surfaces. When the crank chamber was replenished with oil the supply to the cylinder walls was generally excessive and the engine in consequence would pour dense smoke from the muffler. Some makers found it necessary, in order to reduce this smoking, to provide a baffle plate between the cylinder and crank chamber with an opening in the form of a rectangular slot for the connecting rod to pass through. Moreover, the lubrication required frequent attention from the operator, and if the latter was careless the engine could easily be seriously injured through lack of oil. This system, therefore, is now obsolete.

Circulating Splash System—An improvement on the above was the circulating splash system, which is still being used on low-priced passenger cars and some light trucks. Referring to Fig. 268, the oil is carried in a sump or an oil well at the bottom of the crankcase, from which it is drawn by the pump and delivered through a circulation indicator on the dash and through a distributing pipe extending the length of the crankcase, into splash troughs under all of the connecting rod heads. These troughs are provided with overflows as indicated, and the oil therefore is kept at a constant level. The pump may be of any of the three types which will be described further on, and its delivery tube runs to a circulation indicator directly in front of the driver, which at all times gives visual evidence whether or not the oil is circulating.

If the pump is well made there is, of course, little danger of stoppage of the circulation, and in some of the cheaper cars the indicator is omitted.

Each of the connecting rod caps is provided with a splasher or dipper, which dips into the oil in the trough below the connecting rod at each revolution of the crankshaft and splashes the oil over the whole of the interior surface of the crankcase, into the piston and onto the exposed portion of the cylinder wall. Directly in front of the dipper a hole is drilled through the connecting rod cap, and it is intended that oil shall flow through this hole to the crankpin bearing. Whether it actually does so has been questioned, because to do so while the dipper passes through the oil in the trough

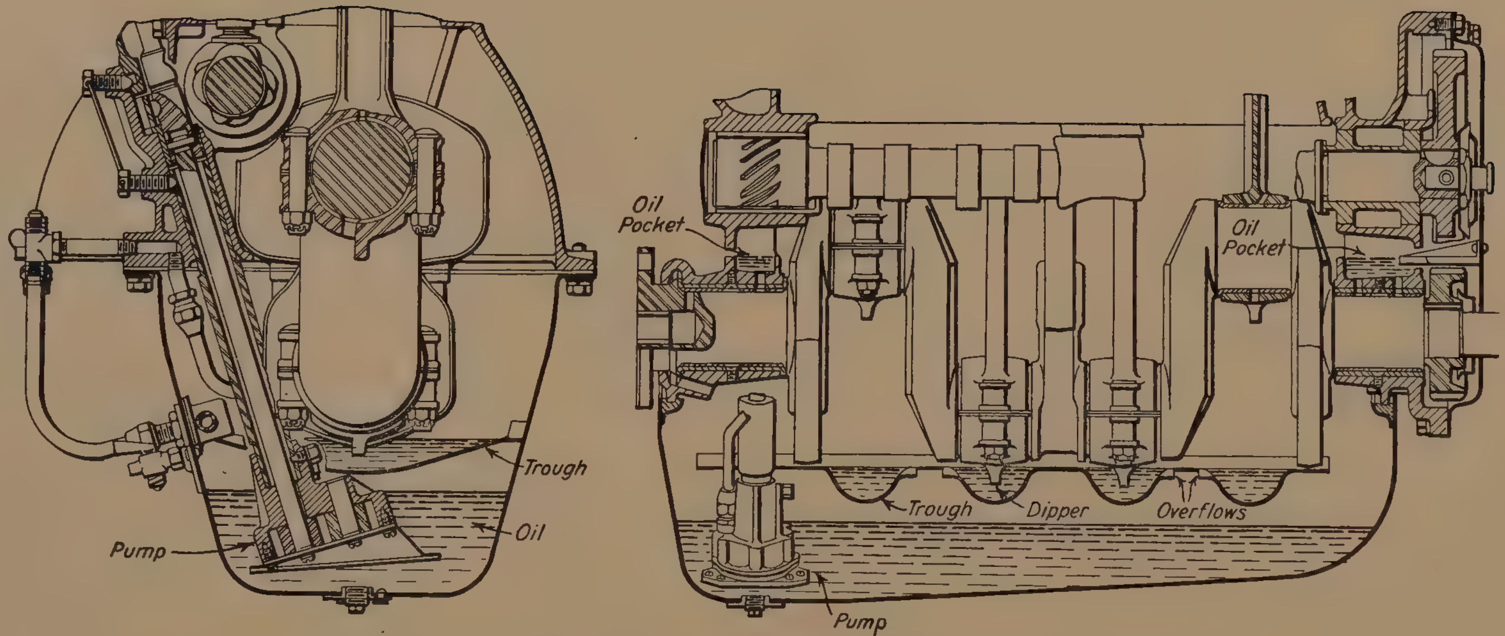


FIG. 268.—DIAGRAM OF THE SPLASH-SYSTEM OF LUBRICATION (NASH).

would mean that the oil was flowing in opposition to the centrifugal force. It is more likely that some of the oil adheres to the connecting rod cap and is forced through the hole to the bearing surface by centrifugal force during the upper half of the revolution. Some makers drill two inclined holes through the connecting rod head on the upper side, or opposite the cap, and such holes are shown in the sectioned connecting rod in the illustration. Some of the oil splashed around in the crankcase will enter these holes and pass to the bearing surface while the crankpin is making the lower half of a revolution, when the centrifugal force is in the direction from the inner side toward the axis of the bearing.

Oil pockets, which catch the splashing oil, are provided over all of the main bearings, and also over the camshaft bearings, and any oil caught in these pockets drains to the bearing surface through holes drilled through the bearing hub and its bushing.

Instead of delivering the oil into the splash troughs, it may be delivered into the pockets over the bearings, by branches from the main distributing pipe. In that case the overflow from these pockets, as well as the oil working through the bearings, and draining down the crankcase walls, collects in the troughs below the crank throws and is splashed around. Another alternative consists in distributing the oil to both the pockets over the bearings and to the splash troughs. With these splash systems, in addition to the leads to the oil pockets or splash troughs, there is generally also a lead delivering into the timing gear housing.

Automatic Circulation in Crank Chamber—Instead of providing a separate feed to each compartment of the crank chamber, some manufacturers endeavor to keep the splash level in the different compartments substantially constant by means of a system of internal circulation (Fig. 269). Oil from the sump is pumped through a sight feed on the dash into one of the crankcase compartments, say the front one. It is splashed around by the crank in this compartment, and the spray draining back down the walls of the compartment is caught in inclined troughs or gutters on the walls, and conducted into the compartment directly in the rear. There it is splashed over the walls again, and again collected in a gutter and conveyed to the compartment in the rear. From the rear-most compartment of the crankcase it is conducted into the sump, whence it is recirculated by a pump through an indicator. In one make of engine the oil circulates in the crankcase in the manner described above, but a pump is dispensed with, the oil finally collecting in the flywheel housing where it is picked up by the revolving flywheel, from which it is thrown off by centrifugal force and some of it is caught in a passage leading to the front compartment.

Splash lubrication is impracticable with V type engines because one set of cylinders would receive much more oil than the other, and either one set would smoke or the other run dry. Cylinders may, however, be lubricated by the oil spray from the ends of the connecting rod bearings, which is thrown equally in all directions.

Pressure-Feed System—The system of engine lubrication most extensively used at present is the pressure system, whereby the oil is forced to the bearing surfaces under a pressure of 30 lbs. per sq. in. or more. When oil is forced to the bearings under considerable pressure, it is more certain to reach all parts of the bearing in adequate quantities. It is generally agreed that with pressure oil feed the specific pressure on the bearing can be made greater, and the bearing will also last longer than with gravity feed. On the other hand, the pressure-feed system is more expensive. This system is to be recommended, however, for all engines whose average load is a considerable fraction of the maximum of which they are capable (high load factor), as well as for all high-speed engines, because of the large bearing loads due to inertia and centrifugal forces in the latter.

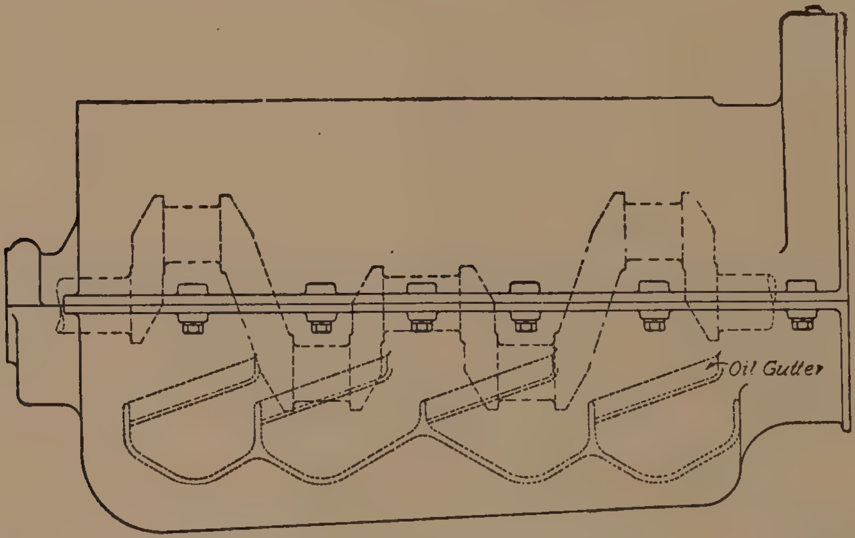


Fig. 269.—Crankcase with Gutters for Automatic Circulation of Oil.

In the force-feed system, of which one application is illustrated in Fig. 270, oil is drawn from the supply in the sump by means of a pump and is forced to the main bearings of the crankshaft through a distributing header with branches to the caps of all main bearings. On the pump there is a pressure relief valve which opens when the pressure in the system attains a certain predetermined value, so that the oil moved, instead of flowing to the bearing surfaces, is returned to the inlet side of the pump.

The crankshaft is drilled from end to end, and from the three main bearings the oil flows under pressure to the six crankpin bearings. From the crankshaft main bearings oil ducts drilled in the crankcase lead to the camshaft bearings, so that these also are lubricated under pressure. From the rear bearing an oil lead extends to the pressure gauge on the instrument board. A groove is turned around the outside of the rear bearing bushing, and a lateral groove extends from it to the point where the oil lead to the gauge enters the bearing.

From the center main bearing an oil lead extends up to the rocker-arm shaft on the cylinder head. An oil hole drilled partly through the center main bearing communicates with the oil hole to the crankpins, and once during each revolution it establishes communication with the oil outlet to the rocker-arm shaft. Thus the feed to this shaft, while positive, is intermittent.

Lubrication of the rocker mechanism is clearly illustrated in one of the detail views in Fig. 270. Oil enters the hollow rocker arm shaft through one of its bearings. The rocker arms are bushed, and inside each bushing there are four small oil holes in the wall of the hollow shaft, through which the oil reaches the bearing surface. The rocker arm has drilled in it an oil hole extending from its bearing to the seat of the adjusting screw, through which the oil is discharged into the cup at the top of the valve operating rod, in which is located the ball head of the adjusting screw. At the other end of the rocker lever the end of the valve stem is submerged in a pool of oil in the conical spring-retaining washer. Oil working out of the various bearings on top of the engine flows through a gutter toward the rear, and returns to the crankcase through the shaft through which the rearmost valve rod passes.

How the oil passes from the crankshaft main bearing to the camshaft bearing is shown in somewhat greater detail in the small insert at the right. At the forward end of the camshaft there is a thrust plunger, and the bearing of this is oiled through an axial and a radial drill hole in the cam-

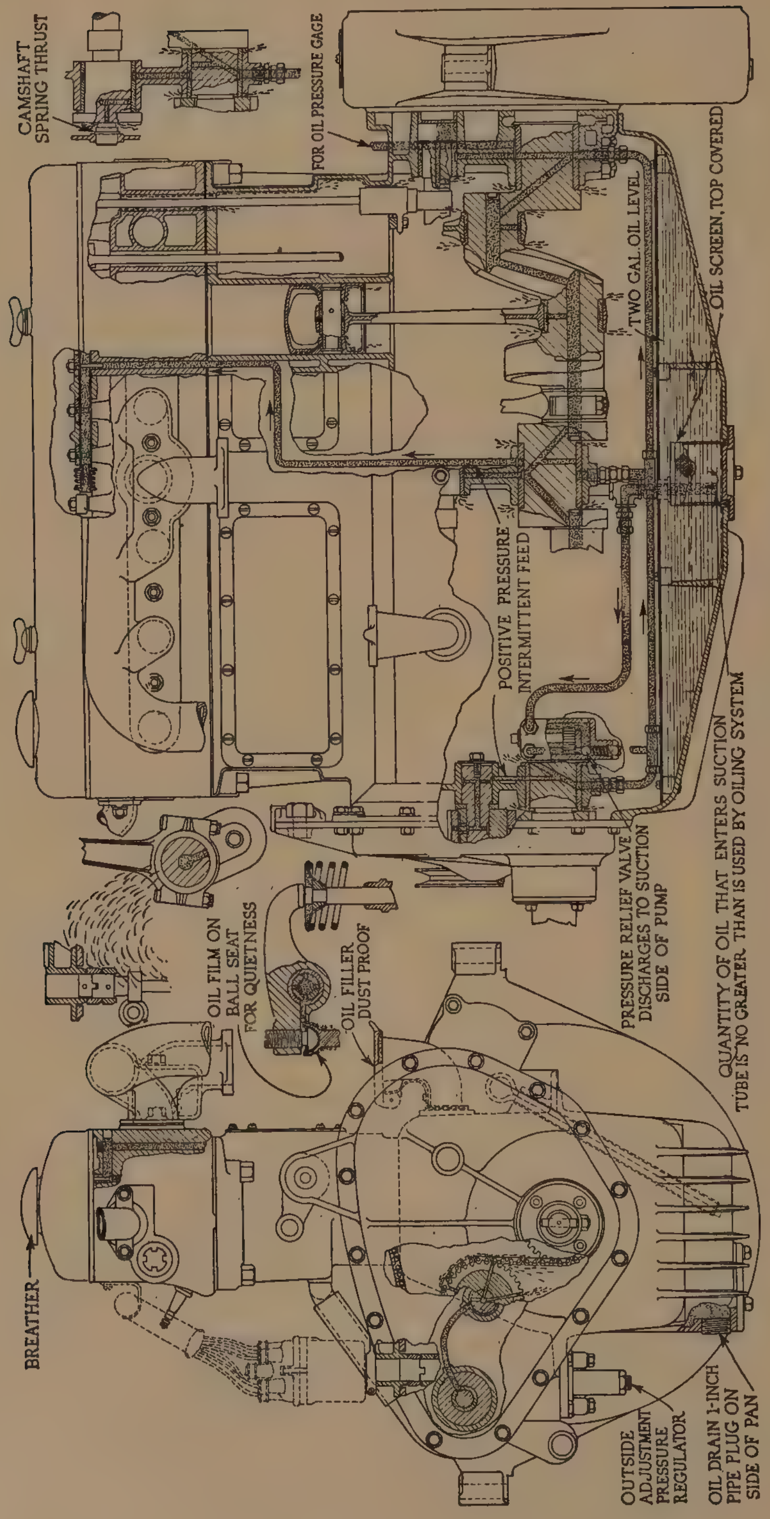


FIG. 270.—FORCE-FEED SYSTEM OF LUBRICATION.

shaft front journal, the radial hole communicating at intervals with an oil groove in the camshaft bearing.

From the forward main bearing there is an intermittent feed of oil to the accessories driveshaft bearing and the idler sprocket bearing. The idler sprocket is drilled with radial holes, and when these come opposite holes in the stud upon which it revolves, oil squirts from them to lubricate the chain. The helical gears by which the oil pump and distributor shaft are driven from the horizontal accessories driveshaft, are directly in the path of the spray from one of the crankpin bearings.

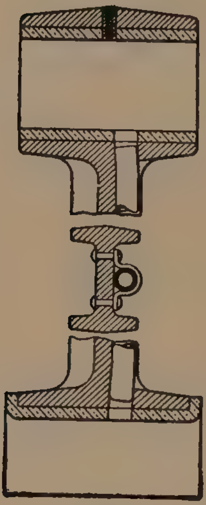
The only important bearings of this engine which are not pressure-oiled are the piston pin bearings. The reason for this is that the pins float, and therefore have bearings in both the connecting rod top end and in the piston bosses, both of which must be lubricated. Oil scraped off the cylinder wall by the ring immediately above the piston bosses flows through two drill holes to the bearing surfaces in the bosses. An oil hole is drilled through the top of the connecting rod to catch some of the oil leaking from the under side of the piston head, and holes are also drilled through the piston pin, so that any oil on the inside of the latter can get to the bearing surface in the small end of the connecting rod.

In this engine the oil sump slopes from both ends toward the middle, where the removable strainer and drain plate are located. The strainer is of cylindrical form and the pump inlet pipe enters it at the top, extending to near the bottom. Two transverse baffle plates or anti-surge plates divide the sump into three compartments. The sump is separated from the rest of the crank chamber by a gauze strainer. The oil filler and the breather or vent are separate devices in this case, the former being located at the side of the crankcase and provided with a dust-tight cover, while the vent is on top of the engine.

In a good many engines the piston pins also are lubricated by oil under pressure. This is the more necessary because the unit pressure on this bearing is higher than that on any other. It is supplied with oil from the crankshaft bearing through a hole drilled through the center of the connecting rod or through a tube passing up alongside the connecting rod shank (Fig. 271).

When oil is fed to the bearings at about 30 lbs. per sq. in. pressure, a good deal works through them and is thrown off the crank arms, etc., onto the interior cylinder, piston and crank chamber walls. This oil is depended upon to lubricate the cylinder, the valve gear, etc. All of the oil thus thrown off, except that which reaches the flame-swept surfaces of the cylinder wall and is burned, returns to the bottom of the crankcase and collects in the oil sump from which it is drawn by the pump.

Relief Valve.—With all force-feed systems there is provided a relief valve, Fig. 272, which automatically keeps the pressure of feed constant, and can be set for different pressures. It consists of a spring-loaded valve in a bypass passage from the delivery side of the pump to the suction side. When the oil pressure exceeds that for which the valve is set the latter opens and allows some of the oil to return to the pump inlet or to the sump, thereby relieving the pressure in the system. Under certain conditions when the engine is new or has had its bearings refitted, it is desirable to run with a higher pressure on the oil, and this can be had by adjusting the relief valve of the regulator. Often the regulator forms a part of the pump.



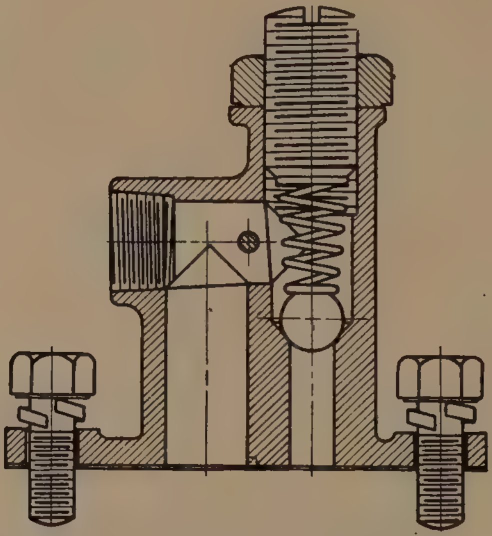


Fig. 271. (left).—Oil Tube to Piston Pin Bearing.
Fig. 272 (right).—Pressure Relief Valve (Wills).

The above described system is the extreme mechanical system in which the largest number of parts receive their lubricant directly by mechanical pressure. From it can be evolved a number of other systems by increasing the number of bearings oiled by the oil spray and correspondingly reducing those oiled by direct force feed. Thus, for instance, instead of feeding the oil from the crankshaft main bearings to the crankpin bearings through oil ducts all the way, the oil working through the main bearings at their inner ends may be caught in oil rings, as shown in Fig. 273, and from these conducted through diagonal holes to the crankpin bearing surfaces. Such a diagonal hole, by the way, is not as objectionable as a radial hole, since a radial hole considerably reduces the amount of stock at a certain section—and that usually at the point of maximum stress—thus tending to concentrate the strain and considerably weakening the shaft. This, however, is of less importance now than formerly, owing to the extremely large diameter crankshafts now commonly used.

Combined Force Feed and Splash—On truck engines a system is used to a certain extent in which a supply of oil is kept in a tank located alongside the cylinders, where the oil will be quickly heated up when starting from cold, so that conditions of lubrication become normal shortly after starting. This is exemplified by the oiling system of Mack truck engines illustrated in Fig. 274. Oil from the sump is raised by a gear pump A located there, through tube B to the oil tank C located alongside the front cylinder block. It is there strained and heated when first starting from cold, but cooled when working under heavy load, as the crankcase oil may then reach 300° F., while the cooling water does not exceed 190° F. From the tank C the oil flows by gravity through tubes E directly to the three main bearings, and through tubes F to the oil troughs G. An overflow pipe from tank C conducts surplus oil to the housing of the helical gear for a cross-shaft for the pump and magneto drive, which housing is part of that for the front end drive. A tube K leads to a pressure gauge on the dashboard. An oil gauge J is provided on the

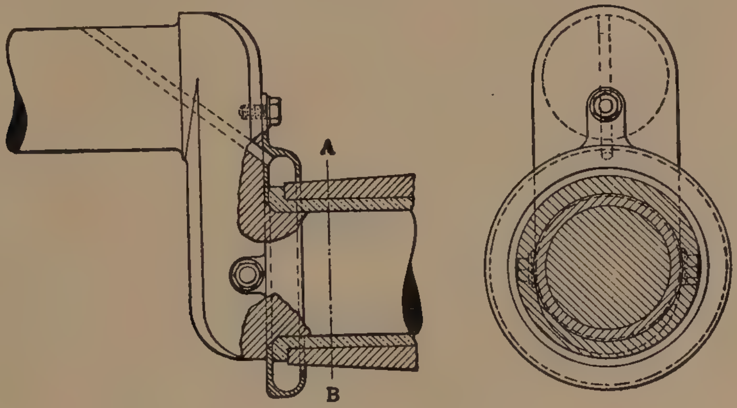


Fig. 273.—Crankshaft Oil Ring.

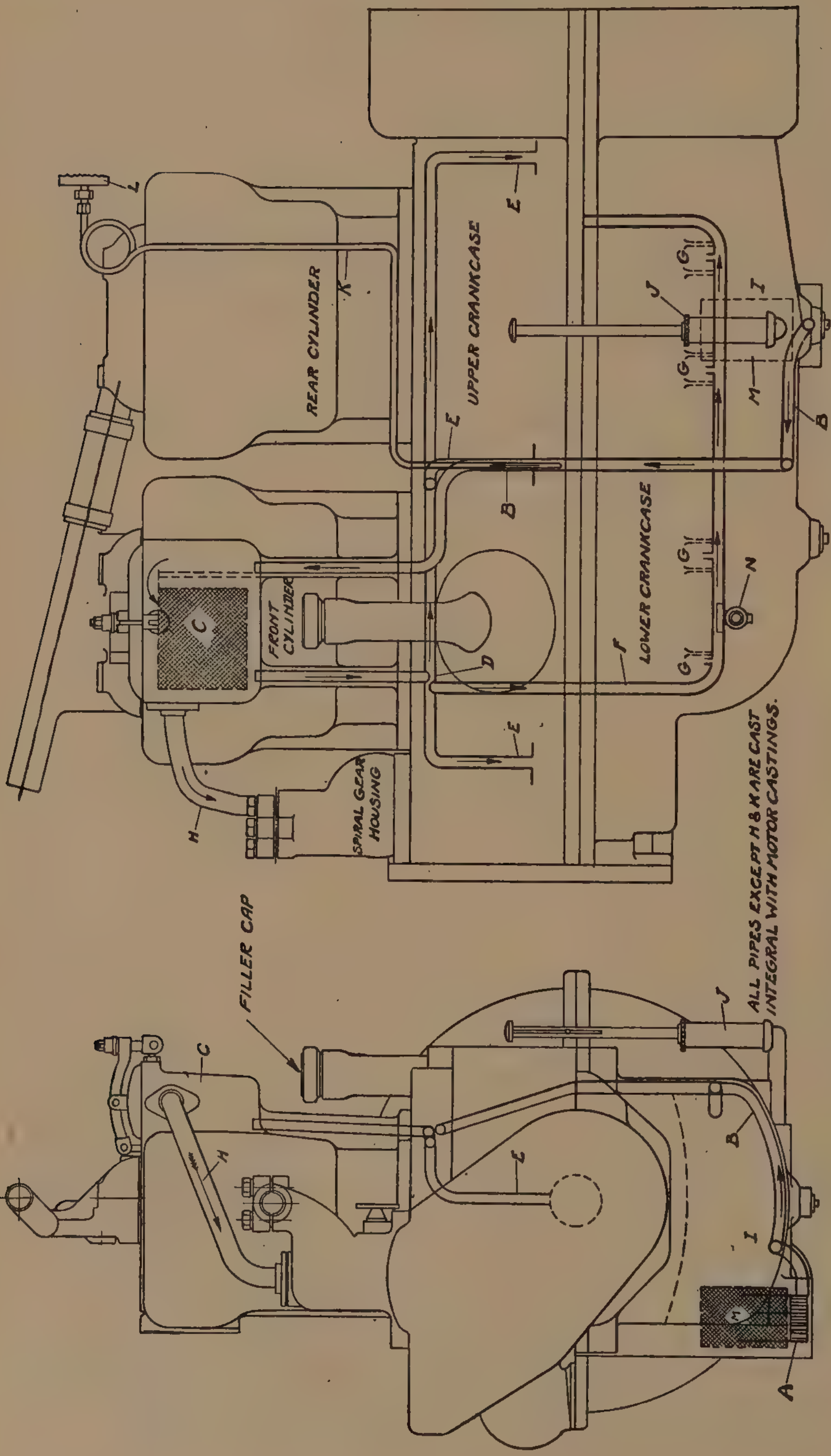


Fig. 274.—Combined Splash and Pressure System with Elevated Oil Reservoir (Mack).

side of the crankcase, and in addition there is a gauge cock N. All of the tubes, with the exception of H and K, are cast in.

Gear Oil Pumps—The most widely used type of oil pump is the gear pump. This consists of a casing in which fit snugly two spur gears, one driven by means of its shaft and the other by meshing with the former. The oil enters the housing on the side on which the meshing teeth separate and fills the spaces between adjacent teeth and the wall of the housing. It is thus carried around with the teeth to the opposite side of the housing and leaves through an opening there. As there is little pressure on the teeth of the gears, and as they run constantly in an oil bath, the wear is almost *nil*. The gear pump is used particularly in connection with force feed systems, in which oil is led to the bearing surfaces under a pressure of, say, 30 lbs. per sq. in. It is capable of giving a high pressure even at a moderate speed, but the pressure under which the oil is fed to the bearing is determined by an adjustable relief valve or by-pass valve.

A good rule in regard to rate of circulation in circulating splash systems is to provide for the delivery per minute of 25 cubic inches of oil per square inch of projected bearing surface supplied directly, at the maximum speed of the engine. Suppose a 3½ × 5-in. engine having a 2-in. crankshaft with three main bearings of an aggregate length of 9 in. The total projected bearing surface then is 18 sq. in. and the pump should move

$$18 \times 25 = 450 \text{ cubic inches per minute}$$

at a speed of, say, 1800 r.p.m.

The delivery of oil by a gear pump may be calculated as follows: Let d be the pitch diameter of one of the gears, p the diametral pitch, h the height of the teeth ($= 2/p$), f the face width of the gears, and n the number of revolutions per minute (which is generally equal to one-half the revolutions of the crankshaft). It may be assumed that the spaces between the teeth are equal in volume to the teeth. The total volume of the annulus comprising the teeth and the spaces is

$$[(d + h)^2 - (d - h)^2]\frac{\pi}{4}f = \pi dhf.$$

The amount of oil transferred from the one side of the housing to the other by one gear in one revolution is evidently half this, and the two gears together transfer a quantity of oil which is represented by the above expression. This is the theoretical delivery. The actual delivery is somewhat less, owing to leakage over the ends of the gears and between the meshing teeth, but in calculating the required size of pump the theoretical delivery can be used as a basis.

Substituting $2/p$ for h and inserting n, the number of revolutions of the pump gears per minute, we have for the quantity of oil delivered:

$$Q = \frac{2\pi dfn}{p} \text{ cubic inches per minute. . . . .} \qquad (91)$$

It will be noticed that the delivery is inversely proportional to the pitch number, hence a relatively small pitch number should be chosen.

If for the pump for our 3½ × 5-in. engine we choose a pitch of 8 and a pitch diameter of 1¼ inches (10 teeth gears), then at 900 r.p.m. the oil delivery will be

$$\frac{2 \times 3.14 \times 1\frac{1}{4} \times 900f}{8} = 884f.$$

and since 450 cubic inches is required we have

$$884f = 550$$

$$f = \frac{450}{884} = \frac{1}{2} \text{ inch (approx.)}$$

Feeding a liberal amount of oil to the bearings is desirable, because the oil, in addition to acting as a lubricant, serves to carry off the heat generated by friction in the bearing, and thus tends to prevent overheating.

As regards the size of the oil pumps used in pressure systems, this does not vary much with the size of the engine. The size required for even the largest truck or passenger-car engine is so small that there is no material gain in reducing the pump dimensions much further in the case of the smaller engines. An average size of gear pump for a large truck or passenger-car engine is one having gears of 1½-in. pitch diameter and ⅞-in. face. For a small engine this may be reduced to 1¼-in. pitch diameter and ⅝ or ¾-in. face.

A design of gear pump for circulating the oil is shown in Fig. 275. The pump consists of the housing, the cover and the two gears. The shaft through which the pump receives its motion extends through its bearing. The idling gear revolves on a stud secured into a hub formed on the housing.

Inlet and outlet openings are on opposite sides. The pump is shown completely surrounded by a screen by which the oil is strained before it can reach the inlet.

Plunger Pump—Another type of pump used for circulating the oil is the plunger type. The plunger is generally forced into the barrel by an eccentric and returned automatically by a spring inside of it or surrounding it. Such a pump may be placed either on the inside of the crankcase underneath the camshaft, or it may be bolted against the crankcase from the outside, in a horizontal position, with its axis intersecting that of the camshaft. A plunger pump to be mounted on the outside of the crankcase is illustrated in Fig. 276. The plunger A is drilled out from one end, and a coiled spring, B, is inserted in it, which holds the plunger against the eccentric. As the plunger moves toward the camshaft, oil is drawn in through the ball check valve C, and during the return stroke of the plunger it is forced through the delivery check valve D, and out through the delivery passage E. Exigencies of manufacture make it necessary to make the delivery check valve much larger than the suction check valve, and to insure quick closing of the delivery valve it is provided with a spring.

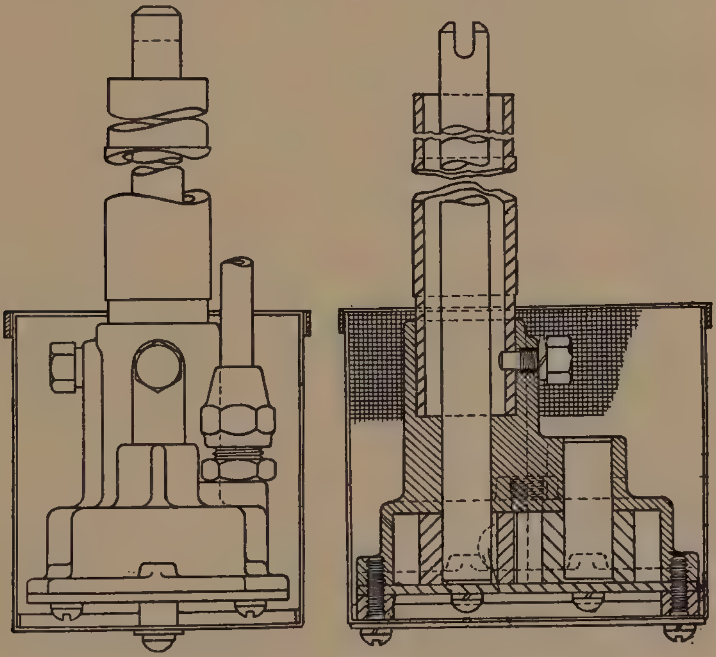


Fig. 275.—Gear-Type of Oil Pump.

Some use has been made also of a type known as a vane pump. This consists of a cylindrical chamber within which is located, eccentrically, a cylindrical block of the same depth as the chamber but only about four-fifths its diameter. This block is supported by a shaft having a bearing in one end wall and is cut into halves by a central saw-cut in which are located two vanes that are forced apart and into contact with the wall of the chamber by a steel spring between them.

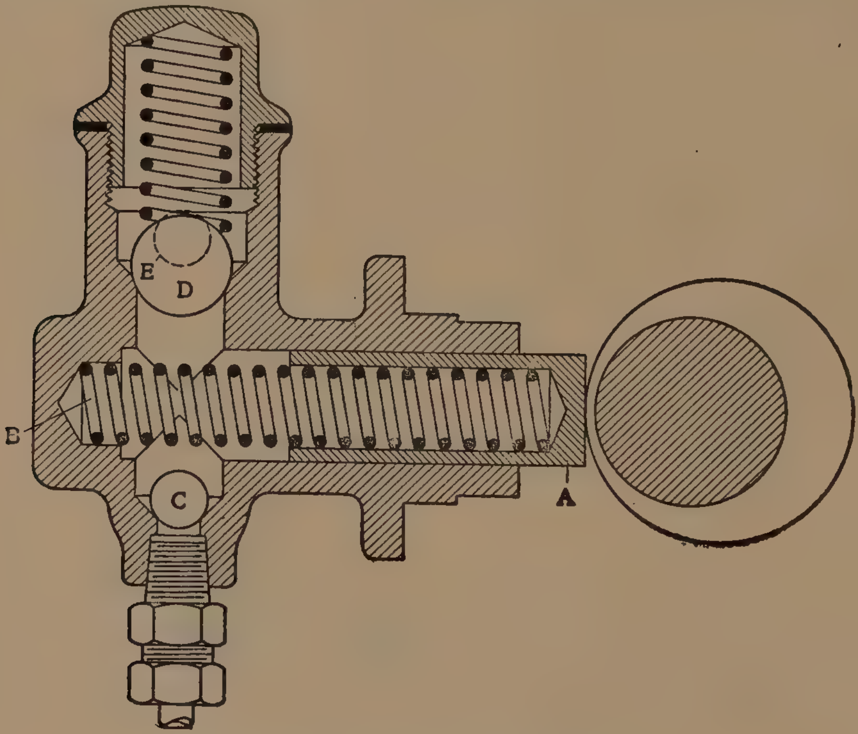


Fig. 276.—Plunger Type of Oil Pump.

Location of Oil Inlets—Oil may enter a bearing either from the outside or from the inside (through the journal). In most instances, of course, it enters through a drill hole in the bearing, but in some cases the oil is fed into a hollow shaft at the end or at one bearing, and passes to the other bearings through holes drilled through the wall of the shaft. In either case the question arises as to the best angular position of the oil hole.

Experiments made by Beauchamp Tower about 40 years ago showed that it is impossible to introduce oil on the loaded side of a bearing carrying a uni-directional load. Tower in his tests employed a bearing which surrounded the journal only half way, resting on top of it and carrying the load. The lower part of the journal ran in an oil bath, and oil adhering to it was carried along and formed a film between the journal and the half bearing. When a hole was drilled through the bearing with a view to introducing oil from the top, it was found that the oil carried up from the bath into the space between the journal and the bearing rose in this hole and overflowed. Tower then drilled holes in the bearing at various definite points, one at a time, and explored the pressure distribution in the oil film over the whole bearing surface, by tapping the holes and inserting a pressure gauge. The total or aggregate pressure on the entire surface naturally is equal to the load on the bearing, and as the distribution is non-uniform, the highest pressure on the oil film is greater than the average unit load on the bearing. The pressure is a maximum near the center of the bearing; it does not change materially along the length, except near the ends, where it drops off very rapidly. Over the projected width of the bearing the drop in oil pressure, from its point of maximum value near the center, follows a curve not unlike a sine curve in appearance.

Direction of Bearing Loads—The load on the piston pin bearing swings with the connecting rod around a central vertical line, but it also reverses in direction near the middle of the inlet, compression and exhaust strokes, and at the beginning of the power stroke (under considerable engine load). Therefore, whether the oil enters the bearing from the inside or the outside, it can be introduced at either the top or the bottom.

The load on the crankpin bearing, being due largely to the centrifugal force on the connecting rod head, is a rotating one, and oil can be introduced into this bearing from the outside at any point of the circumference, each part of which is completely relieved of load at a certain period during the revolution. It is different, however, if the feed is from the inside, as it is in all engines with pressure-lubrication systems. The load on the crankpin revolves in space once during every crankshaft revolution, but the crankpin also makes one revolution during the same time. Hence the load is always on the same side—the inner side—of the crankpin. It swings back and forth slightly, as the load does not revolve with uniform angular velocity while the crank does, but its displacement from the center of the inside is only very small. The inner side being the loaded side, the feed to the crankpin bearing surface must be on the outer side of the pin.

Substantially the same thing holds true with respect to the main bearings. The loads on these bearings revolve in space, and if oil is fed from the outside it may be introduced either on top or at the bottom. But since the journals revolve at the same speed as the load, if the oil comes from inside the journal the hole must be located on that side which is free of load, that is, the side opposite the adjacent crankarms.

Oil Grooves—In order that the lubricant which is introduced into the bearing may be properly distributed over the entire surface, oil grooves are cut in the latter. These should be somewhat different in form according to whether splash or pressure feed is used. However, it is difficult to lay down any definite rules, as opinions regarding the best form of grooving vary widely. In the case of a piston pin bearing which receives its oil through a hole on top, some use a straight groove through the oil hole extending to within, say, ¼ in. of each end of the bushing, as shown at A in Fig. 277, while others use the more elaborate form illustrated at B. This latter groove not only carries the oil close to both ends of the bearing surface, but also all the way around the circumference. In Fig. 277 only the top half of each bearing is shown, and it should be understood that the grooving of the lower half of bearing B is similar to that of the top half.

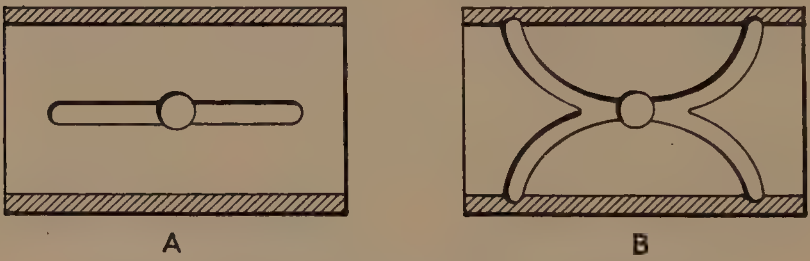


Fig. 277.—Oil Grooves for Piston Pin Bearings.

In the case of main bearings lubricated by splash, the oil always enters at the top, and usually through a single hole at or near the middle of the length. It is then customary to cut oil grooves in the upper half of the bearing only, as shown at D in Fig. 278. In the past it has been customary to provide such bearings with relief at the sides, as indicated in view A. That is, the bearing surface on both sides of the parting lines is undercut to within a short distance of both ends, so that when the bearing is assembled an oil reservoir is formed on each side. This practice probably was taken over from stationary or power plant engineering. If the load on a bearing is always in one direction, there is an advantage in giving the bearing relief at right angles to the diameter in line with the load, because the bearing surfaces thus eliminated add practically nothing to the load capacity, while they add materially to the friction of the bearing, particularly at light loads. In the crankshaft main bearings the bearing load is a rotary one of almost constant magnitude, hence the bearing surface close to the lines of division is as valuable as any other part of the bearing, and this relief is now frequently omitted. When relief is provided it is in the form of slots covering one-half or less of the length of the bearing, as indicated at C, Fig. 278. These slots serve as oil reservoirs.

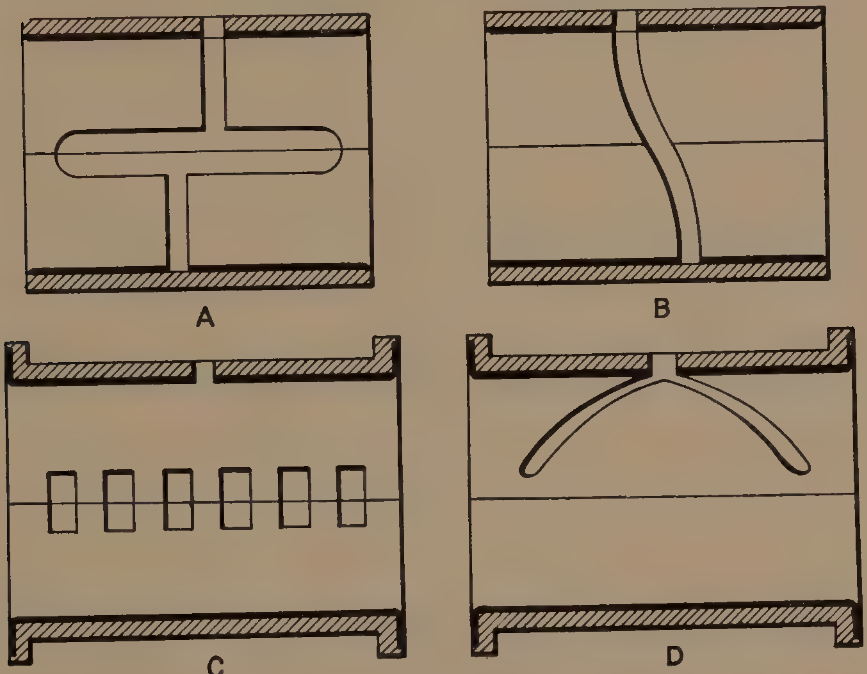


Fig. 278.—Oil Grooves for Main Bearings.

A fact worth remembering in this connection is that toward the edges of any bearing surface—and an oil groove communicating with the oil-feed inlet forms such an edge—the pressure in the oil film reduces to zero or to the pressure of the oil feed, which latter is comparatively low. The farther from the edge any particular portion of the bearing surface is located, the larger its share in sustaining the load, and from this point of view it would seem a good plan to omit oil grooves altogether, as they interfere with the "building up" of pressure in the film. However, the oil must be distributed over the bearing surface. Besides, as the pressure in the film increases, the thickness decreases, and if the film becomes too thin there is metallic contact and consequent rapid wear. The minimum permissible film thickness, (which for a given lubricant at a given temperature corresponds to the maximum load), depends upon the accuracy and perfection of finish of the bearing surfaces.

With pressure feed to the piston pin a system of grooving similar to that shown at B, Fig. 277 for splash feed may be used. The oil inlet, in this case, of course, is at the bottom, and owing to the pressure behind the oil it is not necessary to carry the grooves all around the bearing. The simple lengthwise oil groove is not adapted to this case; rather should the groove be cut circumferentially around the bearing near its middle, so the oil can flow from it toward both ends. A circumferential groove, however, should not be cut in one plane, as that would cause a ridge to wear on the pin.

The same applies to main bearings for force feed lubrication. The oil groove, if at right angles to the axis of the bearing, should be in two sections which are staggered with relation to each other. An equally good plan is to have the groove of elliptical form, with its plane inclined to the axis of the bearing. A number of suggestions for oil-grooving main bearings are given in Fig. 278. Designs A and B are applicable to bearings with forced lubrication, while D is intended for a bearing with splash lubrication.

Oil grooves running circumferentially around the bearing are generally made of rectangular section, as they are cut with a milling cutter of the same diameter as the bottom of the groove. Those extending lengthwise are made with a section forming a circular segment. All edges at an angle to the direction of motion should be slightly chamfered so they cannot possibly scrape the oil from the journal. This includes the edges of the bearing bushings and the edges of any oil grooves extending lengthwise of the bearing.

Since the wide application of pressure lubrication there has been some trouble from scoring or scuffing of pistons, particularly aluminum pistons. This is ascribed to the fact that with this system of lubrication, when an engine is started from cold at low temperature, it takes a considerable time before any oil reaches the cylinder walls. Frank Jardine of the Aluminum Co. of America, who investigated this matter, found that the clearances of connecting rod bearings, both diametral and end, were so small that practically no oil could get through as long as the engine was cold and the oil therefore very viscous. This scuffing was entirely eliminated by cutting a groove about 3/16 in. wide and 1/16 in. deep across the face of the connecting rod bearing about 15° back (toward the thrust side of the engine) of a plane through the centerline of the rod and the crank. At the ends of this groove small vertical slots about 3/16 in. wide and 1/16 in. deep were cut in the sides of the bearing. See Fig. 279A.

Oil was found to collect in these side slots and to be thrown off. This arrangement was successful in curing all except the most stubborn cases, in which it was necessary to cut a 3/16 × 1/16-in. circumferential groove or grooves in the connecting-rod bearing, connecting with the cross groove and placed so as to register with the oil hole in the crankpin, as shown in Figs. 279B and C. Mr. Jardine drew the conclusion that while the pressure system of lubrication is undoubtedly the best for the bearings, the splash system is best for the cylinders and pistons, and he advocated a combination of the two systems.

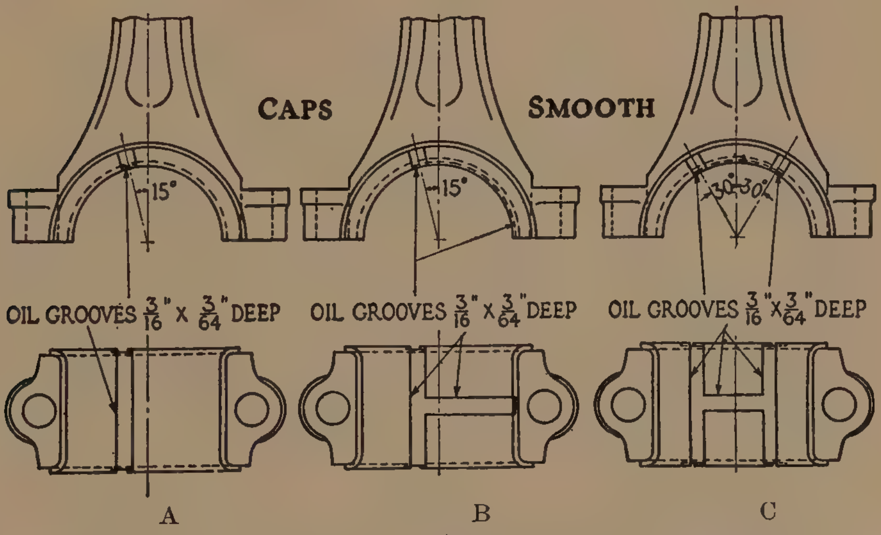

CAPS
SMOOTH
15°
15°
30° 30°
OIL GROOVES 3/16" X 3/64" DEEP
OIL GROOVES 3/16" X 3/64" DEEP
OIL GROOVES 3/16" X 3/64" DEEP
A
B
C

Fig. 279.—Oil Grooves for Connecting Rod Bearings.

Oil Jet to Cylinder Walls—Where pressure lubrication to the crankpin bearing is used, a hole is sometimes drilled in the bearing which registers with the oil hole in the crankpin when the crank is in and near the top dead center position. At that moment a jet of oil is thrown from the connecting-rod head, but while it has been assumed by some that it is thrown off radially (into the cylinder), it is the author's contention that it is thrown off tangentially.

Connection to Main Bearings—There are several methods of feeding oil under pressure to the main bearings. One consists in casting a steel distributor tube into the crankcase wall near the top thereof, with brazed-on branch tubes extending through the bulkheads of the crankcase. This is illustrated in Fig. 280, which shows the crankcase of a Continental engine. The more conventional way, however, is to provide an oil manifold, the header of which extends along the crankcase wall on the inside and the branches of which connect by couplings to the bearing caps. Three different types of connections or couplings used for this purpose are illustrated in Fig. 281.

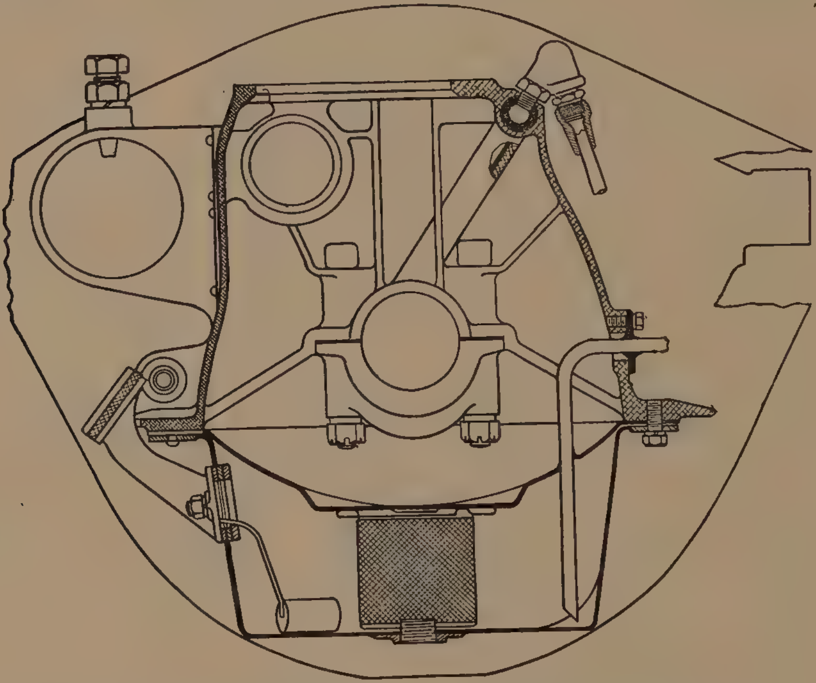


Fig. 280.—Oil Distribution through Oil Tubes Cast in Aluminum Crankcase (Continental).

Continuous Replenishment of Crankcase Supply—With most engines, oil is added to the supply in the crankcase at intervals, usually a quart at a time, and at longer intervals (1000 to 2000 miles of driving) the entire supply is drained off and new oil put in. The objection to this method is that the qualities of the lubricant in the crankcase vary continually, particularly its viscosity, upon which its lubricating value depends to a large extent. Some Continental manufacturers have sought to overcome this effect by providing two pumps, one of which circulates the oil or forces it to the bearing surfaces, while the other, smaller one adds new oil to the

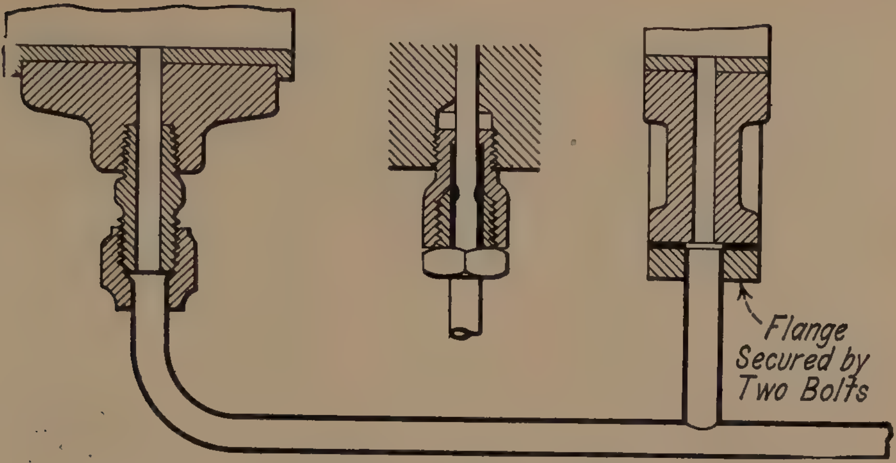


FIG. 281.—OIL-MANIFOLD FOR MAIN BEARINGS, SHOWING DIFFERENT TYPES OF JOINT.

crankcase supply from a tank carried on the engine, at a slow rate. Usually the two pumps are combined in one.

The system used on the Swiss Saurer truck is illustrated in Fig. 282. The pumps are of a special type, the movable part having a combined rotary and reciprocating motion and performing the functions of not only a pump plunger, but also of a valve. An extension of the double plunger near its upper end carries a helical gear, which meshes with a similar gear on the camshaft. This, of course, imparts a rotary motion to the plungers. Near the lower end, and directly adjacent to the larger one of the plungers, there is an enlargement in which is cut a cam slot. A roller on a pin secured in the wall of the pump barrel is lodged in this slot, and the result is that as the plungers are revolved by the helical gears, they are forced to reciprocate at the same time.

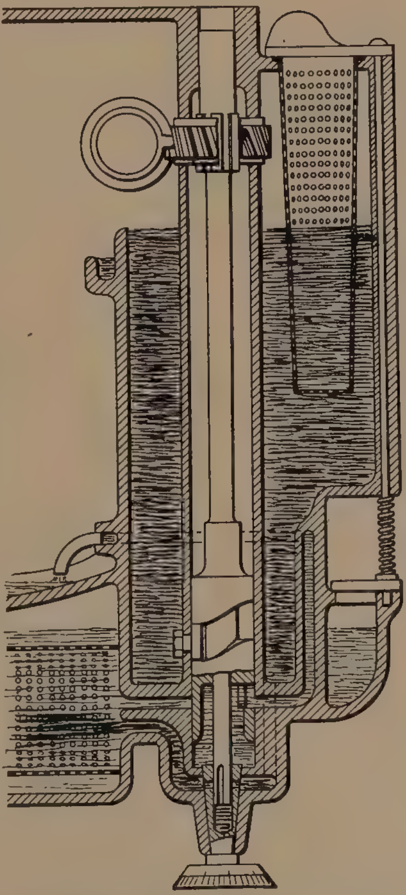


FIG. 282.—CONTINUOUS ADDITION OF FRESH OIL (SAURER).

Each of the two plungers has a port in its wall or face, which registers with the suction or inlet port during the up-stroke and with the delivery port in the barrel during the down-stroke. The small diameter plunger draws oil from the fresh oil reservoir and discharges it into the crankcase sump,

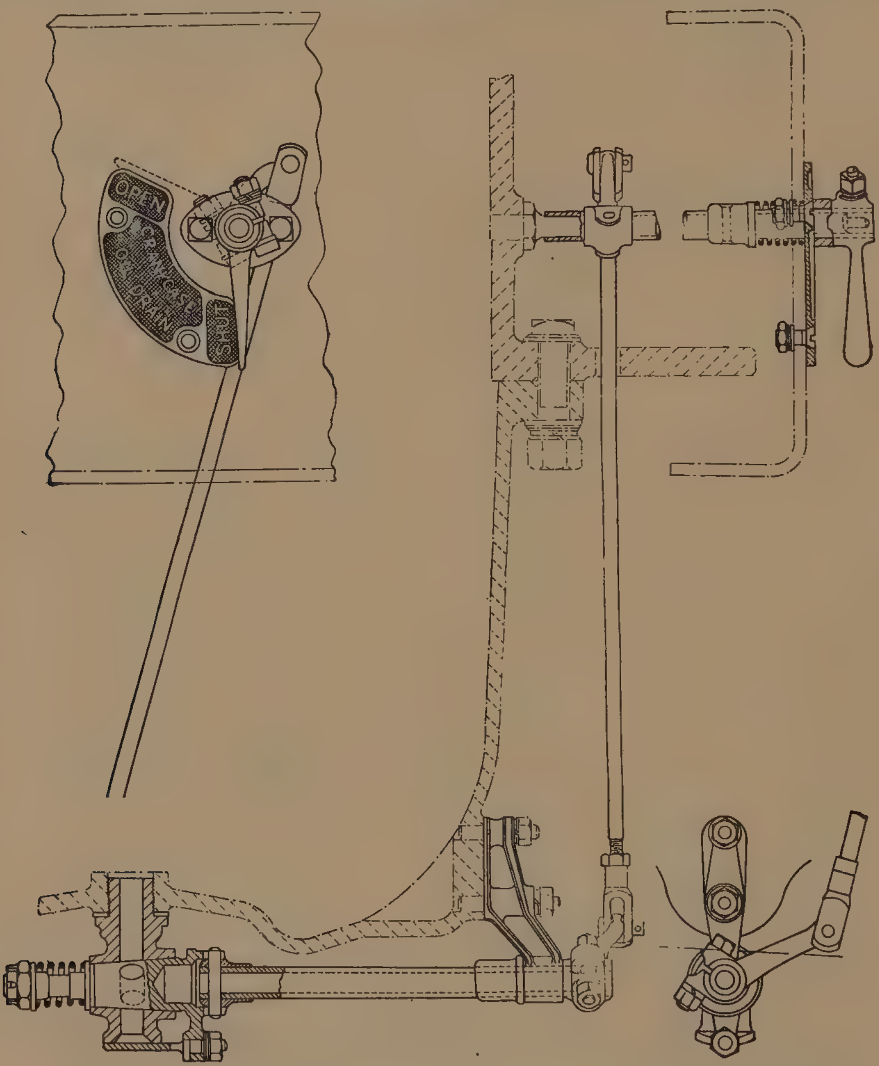


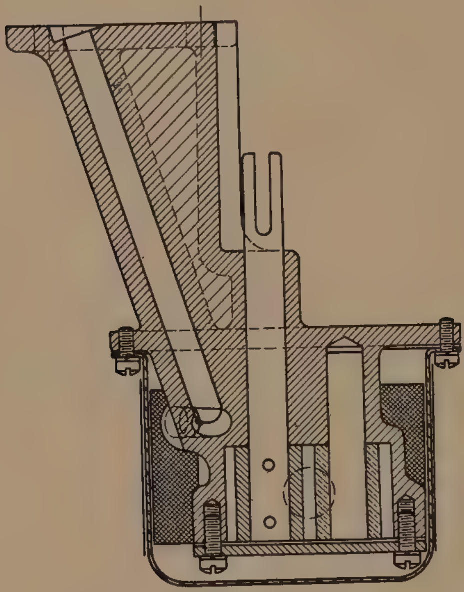
Fig. 283.—Convenient Crankshaft, Drain Valve (Rolls-Royce).

while the large plunger draws oil from the sump and delivers it into the splash troughs. Suction and discharge passages of both pumps are clearly shown in the drawing. The rate of fresh oil delivery can be adjusted by means of the micrometer screw at the bottom of the pumps. An interesting feature of the design is that when the filler cap is swung into the ''open'' position, the cap of an overflow (gauge) opening is swung open at the same time.

Convenient Drain Valves—In practically all of the older and some of the newer engines, when it is desired to drain the crankcase of oil, it is necessary to crawl under the car and unscrew a pipe plug or a number of cap screws, in the course of which operation clothing is often ruined by dirty oil. It is now becoming the practice to provide drain valves which can be operated from above. The somewhat elaborate but most convenient drain valve on the Rolls-Royce engine is shown in Fig. 283. The operating handle is located on the outside of the car frame and moves over a sector plate having ''Open'' and ''Shut'' marks cast upon it. In one American car a valve body is riveted to the pressed steel oil pan near the bottom. This has two adjacent chambers, one open to the oil pan, the other to the atmosphere. In the wall between these two chambers there is a conical seat, which can be closed by a valve plug screwed into the upper part of the valve body and having an extension stem, the upper end of which is shaped to form a convenient handle located just inside the car frame. In another engine of prominent make the crankcase is drained by means of a conical valve which is held to its seat by a coiled spring and is opened by merely pressing on a conveniently located knob.

Method of Mounting Pump—In engines with force feed lubrication, in which the oil distributing tubing or channels are cast in the upper section of the crankcase, it is customary to cast the body of the gear pump with a supporting bracket by means of which it can be bolted from underneath against a pad cast in the top section in such a way that the pump delivery channel extending through this bracket will register with the opening of the oil distributing channel in the top section of the case and the shaft of the pump will engage by means of a slot and key with a vertical pump drive shaft in the upper section of the case, driven from the camshaft by a pair of helical gears. The whole pump in that case is surrounded by a wire screen. Although located in the oil sump, the pump is entirely independent thereof and is not disturbed by the removal of the sump. The construction is shown in Fig. 284. A scheme for driving the oil pump from the rear end of the camshaft through a Hookham joint is illustrated in Fig. 285.

Recently oil pumps in increasing numbers are being placed on the outside of the crankcase, where they are more accessible. Another reason is that in cold weather a good deal of water of condensation is likely to accumulate in the sump, which may cause the pump to freeze up, with the result that it or its drive shaft will be broken when the engine is started up. To prevent this accumulation of water (mainly from exhaust gases) it is now proposed to ventilate the crankcase, either by the pumping action of the pistons or by connection to the inlet system. Water in the crankcase is also very objectionable because it promotes rusting of the steel parts therein, especially during periods of non-use.



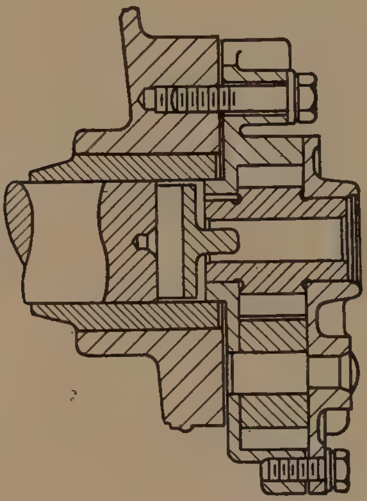
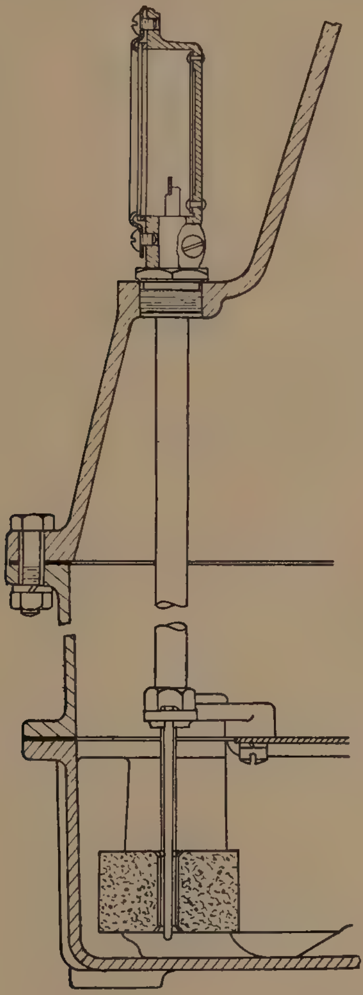


Fig. 284 (left).—Pump Mounted Inside Crankcase.
Fig. 285 (right).—Pump Driven from End of Camshaft

Oil Level Gauge—When the supply of oil is carried in an oil well at the bottom of the crankcase some means must be provided to enable the operator to determine quickly how much remains in the well. Some manufacturers merely provide two try cocks in the side of the oil reservoir, one of which is placed at the lowest level which the oil should be allowed to reach. Others place a bull's eye or glass disc in the side of the well. Both these methods are primitive and now rarely used.

The most satisfactory form of oil gauge comprises a float in the sump from which connection is made to an indicating device outside the crank chamber. Usually a rod extends straight up from the float through a hole in the crankcase wall, sometimes into a glass tube set into a fitting screwed into the upper part of the crankcase. The rod carries a ball or other easily visible device at the top, and the exposed length of the glass tube denotes the permissible range in the oil level in the crankcase. An oil gauge of this type, but with a somewhat differently constructed indicating portion, is shown in Fig. 286. In another form of oil level gauge in extensive use a float in the sump actuates the pointer of a dial gauge through a lever arm extending through the wall of the oil pan. This type may be seen fitted to the crankcase illustrated in Fig. 280.



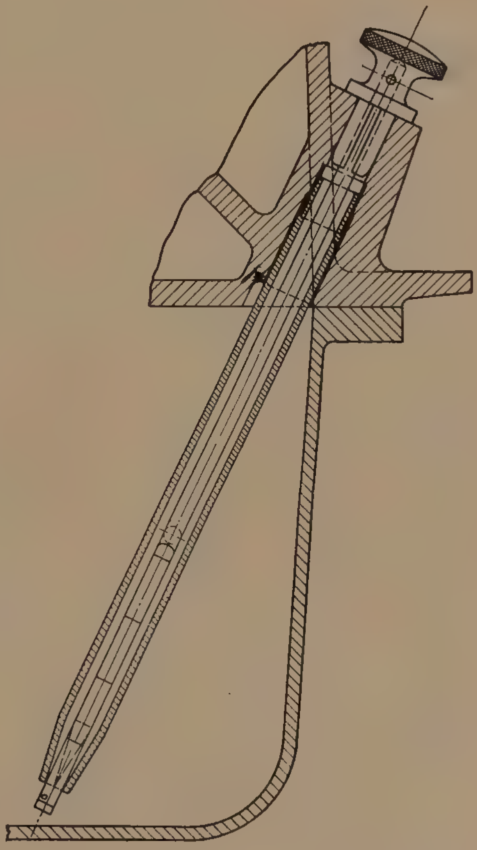


Fig. 286 (left).—Direct-Reading Oil Level Gauge.
Fig. 287 (right).—Spear-Type of Oil Level Gauge.

There is still another type of oil gauge, somewhat cheaper than a high grade float indicator. It is known as the spear type and consists merely of a ¼ in. (or so) steel rod extending down a tube set into the upper section of the crankcase, usually at an angle of about 30° to the vertical. The tube extends to within a very short distance of the bottom of the sump. It is contracted at the bottom so the rod fits it snugly and at the upper end the stem of the knurled knob is a snug fit in the hole, so the gauge will not shake or rattle. At the lower end the rod is flattened off on one side to allow the oil to enter the tube and graduations are marked on the rod showing when the sump contains ¼, ½, ¾ and 4/4 or the maximum amount of oil that should be put in. Such a gauge as used on the Class B military truck engine is shown in Fig. 287.

Lubrication Control in Proportion to Load—With both splash and pressure-feed lubrication an excess of oil is likely to get onto the cylinder walls when the engine is running under light load; the delivery of the pump in a pressure system and the splashing of the oil in a splash system increase substantially in proportion to the speed, but owing to the high suction vacuum in the cylinder at light loads (when the throttle valve is nearly closed) the oil tends to be drawn into the combustion chamber and to cause a smoky exhaust. In a circulating-splash system the delivery of oil to the cylinder walls can be reduced by reducing the delivery of the circulating pump. With a spring-returned plunger pump, for instance the stroke of the plunger can be varied by means of an eccentric or lever which is connected to the throttle valve and which does not allow the plunger to return all the way when the throttle is partly closed.

In pressure lubricated engines there is probably even more trouble from over lubrication at light loads. The pressure relief valve insures that a predetermined pressure on the oil cannot be exceeded when the engine runs very fast, but while this maximum pressure is suitable for full load operation it is too great for light loads. Therefore, the by-pass should be opened somewhat if the engine is running light at considerable speed. This can be effected either by connecting the by-pass with the throttle valve or else by controlling it by the suction in the inlet manifold. The latter plan is used on the Marmon engine.

The Marmon regulator is shown in Fig. 288. It consists of a cylinder A containing a piston B to which is connected a piston valve C. Piston B is chambered out and the coiled spring D, which tends to force it down in the cylinder, extends into B. From the top of cylinder A a small tube leads to the intake manifold, whereby the vacuum in the manifold is communicated to cylinder A.

Port E at the lower end of the regulator communicates with the oil-distributing system. Directly above it is the pressure relief valve F which can be adjusted by means of set screw G. If the engine is being operated under full load at high speed, the pump pressure will exceed the pressure for which the relief valve is set, the valve will open and some of the oil will pass through it and be discharged in the valve chamber on top of the engine, whence it drains back to the oil pan. On the other hand if the engine runs fast under no load or light load, the suction in cylinder A will draw piston B and valve C up against the pressure of spring D and oil from port E will be by-passed through port H back to the sump. Therefore, when the load on the engine drops below a certain value, almost all pressure is removed from the circulating oil.

In the Marmon system all of the oil delivered by the pump enters the hollow crankshaft at the rear main bearing; whatever does not work out through the main and crankpin bearings leaves the crankshaft at the front main bearing through a lead to the regulator. Naturally, the amount of oil working out through the different bearings is dependent upon the pressure on the oil and not upon the rate at which the oil circulates through the hollow crankshaft. However, rapid circulation of the oil tends to cool the bearings.

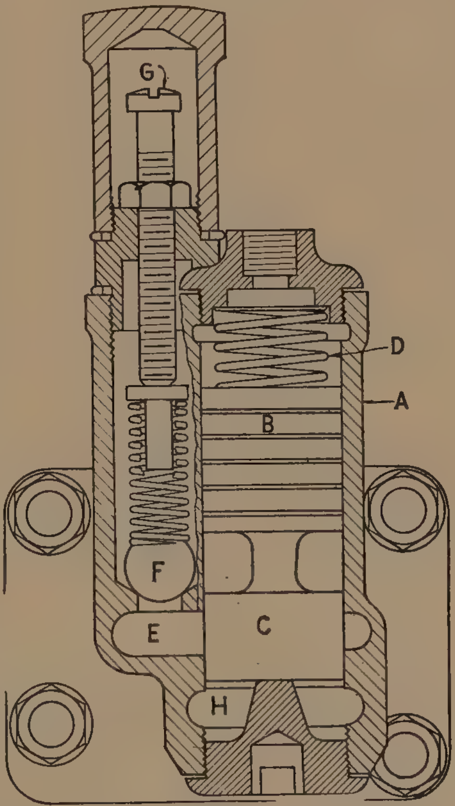


FIG. 288.—MARMON OIL-PRESSURE REGULATOR.

Circulation Indicator—A continuous supply of lubricant to its bearing surfaces is essential to the proper operation of a gasoline engine. If the oil feed stops it is usually only a short time until destructive heating and cutting of the bearings begins, and there should, therefore, be some device constantly indicating to the driver that the oil is being fed properly. If oil is supplied to the bearings under pressure a pressure gauge or indicator on the dash will serve the purpose, since the oil feed could only stop if the pump should cease to work or an oil lead should break, in either of which cases the pressure in the system would become *nil.*

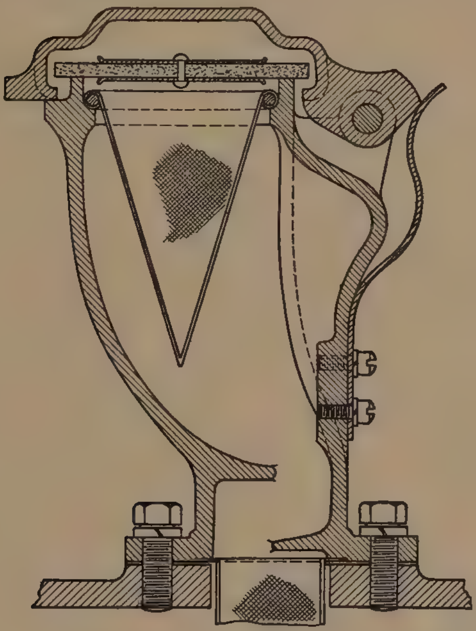
In the case of the ordinary circulating systems the general practice is to carry either all the oil or that fed to one bearing or one crankcase compartment, to the dashboard, where it passes through a circulation indicator. All instruments for passenger-car engines must be of the flush-mounted type, and a suitable circulation indicator consists of a cylindrical metal chamber with glass front, inside of which there is a stamped metal wheel with blades like an air motor, which is kept in rotation by the oil flowing through the chamber.

Oil Distributor—In practically all engines with pressure lubrication the oil is fed to all of the main bearings in parallel. One objection to this method is that if one bearing should become very loose, most of the oil would flow to it, while another bearing might be starved. To obviate this, oil distributors were formerly used to a certain extent, which place the different bearings in communication with the pump in succession. They consist merely of a cylinder revolving inside a housing, with a central port through which the oil from the pump enters, and with one or more ports at or near the circumference which come into register with ports in the housing, from which latter the distributing tubes extend to the bearings. Such a distributor is being used on the Mercedes engine at present, the device forming an integral part with a double pump.

Oil Fillers and Breathers—The oil filler and crankcase breather can be conveniently combined into a single unit, and an excellent design is illustrated in Fig. 289. This was first used on the Class B truck engine. At the bottom there are two baffles which have an oil-separating effect. A strainer of funnel shape, which can easily be removed for cleaning, is inserted from the top. Above this strainer there is a weighted leather disc serving as a check valve and tending to maintain an under-pressure in the crankcase. The filler, moreover, is provided with a hinged cover, so designed that a spring holds it firmly in either the open or closed position.

Rather primitive breathers are used on most of the cheaper passenger-car engines. Often a length of steel tube is used, which is pressed into a boss on the wall of the crankcase. Inserted into it are two steel pressings serving as baffles, and a snap cover fitting the top prevents dirt falling into the crankcase. Some designers prefer to mount the breather on the timing gear case, where it is higher up and therefore in a more convenient location for adding oil. Such an installation (Hupmobile) is shown in Fig. 290.

Size of Oil Conduits—Tubes through which the oil is fed under pressure need not be very large, as the flow through the bearings naturally is at a slow rate. For the manifold distributing oil to the main bearings 5/16 in. outside diameter tubing is largely used. On the other hand, all oil holes and passages through which the oil flows merely by reason of its own weight must be of liberal size, because in cold weather the oil becomes very viscous. One-quarter inch should be the minimum size for such oil holes in connecting-rod bearings, etc., whereas in the case of large truck engines with splash lubrication, the oil holes in the main bearings at the bottom of the pockets should be ⅜ or 7/16 in. The receiving ends of such oil holes are often chamfered out.



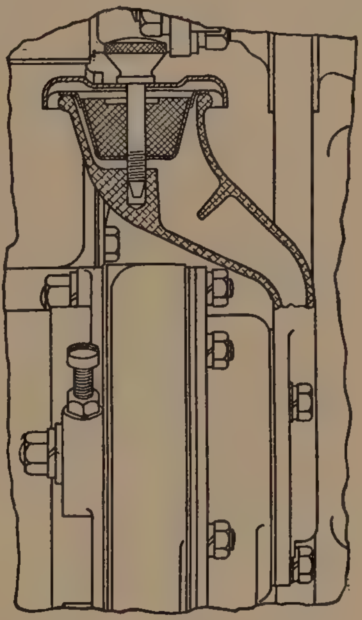


Fig. 289 (left).—Oil Filler with Breather Valve and Strainer.

Fig. 290 (right).—Oil Filler Cast on Timing Gear Case Cover (Hupp).

Oil Strainers—The oil drawn in by the pump must be strained, so that any metal dust worn off the bearings may be separated, because if it should get into the bearings it would start cutting. The strainer should be made of brass wire gauze of about 16 mesh, of such form that it can be easily removed for cleaning. It is also well to remember that as the oil becomes dirty it will clog up some of the meshes, and in order that the flow of oil through the strainer may not be appreciably impeded it should be made of large area. One design of strainer suitable for use in connection with pumps located outside the sump is shown in Fig. 291.

Sludge Valve—When an engine is at rest the heavy particles normally held in suspension by the oil will settle at the lowest point of the crank chamber, where the drain plug or cap is usually located, forming a sediment or sludge. With the ordinary construction of drain plug it is impossible to remove this sludge without at the same time draining off the oil. A valve which enables the operator to conveniently remove the sludge at intervals is shown in Fig. 292 and is of English origin. A sludge pocket with a seat for a poppet valve at its bottom screws into the oil sump at its lowest point. The cap on the sludge pocket when screwed home holds the valve off its seat so that the sludge can settle into the pocket. Before the cap is fully unscrewed the valve reaches its seat and thus prevents oil from draining out of the sump when the cap comes off.

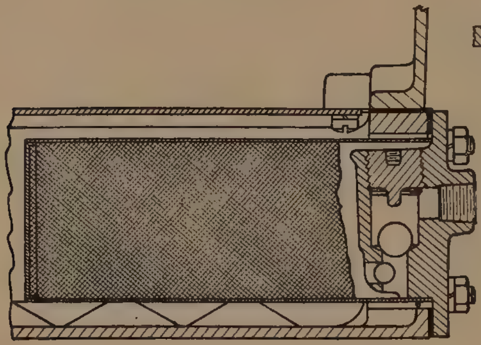


FIG. 291.—OIL STRAINER.

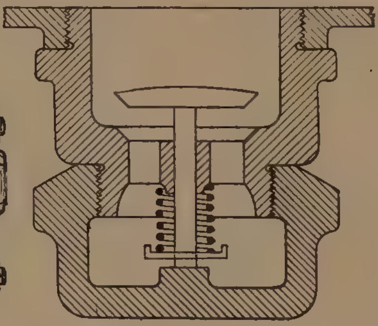


FIG. 292.—SLUDGE VALVE.

Oil Filters and Distillators—The life of an engine depends largely upon its lubrication, and it is therefore important that a proper quality of oil be used, that it be fed to all bearing surfaces in adequate quantities, as long as the engine is running, and, lastly that it be kept in proper condition. There are two chief causes for the deterioration of the oil. One is that solid particles, such as road dust, drawn in through the breather or with the carburetor air; metal particles worn off the bearing surfaces, and carbon particles formed by the coking of the oil on the under side of the piston head, collect in it; the other, that some of the "heavy ends" or least volatile fractions of the fuel pass by the piston, mix with the oil and dilute it, whereby its viscosity and its lubricating value are lowered. Burnt gases also blow by the pistons, and the steam in these gases condenses. The strainer separating the sump from the crankcase and that over the pump inlet are depended upon to a certain extent to keep solid particles out of the circulating oil, but they are not sufficient for the purpose, for, owing to the low pressure behind the oil as it passes through these strainers, they must be made of comparatively coarse mesh and do not separate out the finer solid particles. Therefore, in recent years there have been developed a number of oil filters which are mounted outside the engine and which contain fibrous material (asbestos) through which the oil is forced by the pressure of the pump. Either the fibrous filtering material or the entire container can be renewed when the material has become saturated with sludge.

There also have been developed stills or distillators for removing the fuel and aqueous moisture from the oil. The earliest device of this type was the Skinner oil rectifier, which draws the excess oil from around the pistons and subjects it to the heat of the exhaust, which vaporizes any fuel contained in it, and returns this fuel to the inlet system, while the purified oil is allowed to drain off to the crankcase. With this system, which is now used by Packard, pistons of special design are required. Other distillators draw oil from the crankcase and subject it to distillation by exhaust heat. The principles of the filter and distillator are combined in the Marmon oil purifier, of which a sectional view is shown in Fig. 293. One advantage claimed for the combination of the two appliances is that solid particles are more readily removed from the oil when it is hot, as it is then less viscous, and therefore can be forced through a much finer filter.

The purifier is bolted against the exhaust manifold at the rear of the engine, and the exhaust gases from the two rear cylinders pass through the jacket J surrounding the distillation chamber A. Below the latter there is a filtration chamber B. Four tubes lead to the purifier. Tube K (indicated diagrammatically) is the oil-feed tube from the pressure regulator; E, an air-inlet tube, leading from the crankcase up through the center of B into the upper part of A, venting the distillation chamber when oil is being drained from it; G is a syphon which automatically drains off the oil from A each time the float D rises. The asbestos filtering sack C prevents abrasives from being fed back into the oil. I is an air valve operated by float D. The filter sack C has a very large area and the passage of oil through it is consequently very slow. In the process of filtration, all carbon suspended in the oil is removed. Tube F from the upper chamber A carries fuel and water vapors to the carburetor intake to be drawn into the engine along with the mixture.

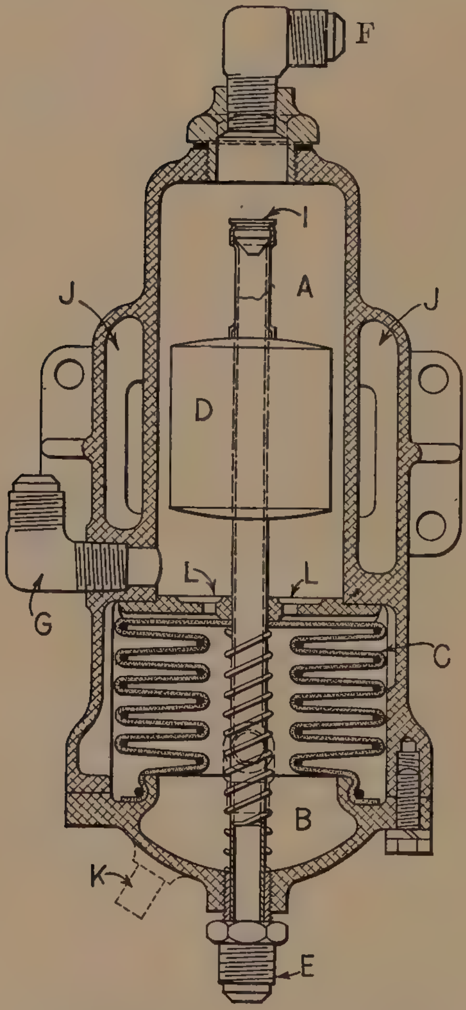


Fig. 293.—Marmon Oil Purifier.

After having passed through the filter sack, the oil passes out of the lower chamber B into chamber A, through the opening L. Here it is subjected to a temperature high enough to evaporate the water and fuel, but not sufficiently high to break up or coke the lubricating oil.

As the temperature of the oil in chamber A increases, any water in emulsion is turned to steam and is drawn off through the vacuum lead F to the carburetor inlet. At a still higher temperature any fuel contained is distilled and passes off. The lubricating oil continues to rise in chamber A until it reaches a level almost even with the top of the float D.

Tube G is filled with oil and when the lubricant in chamber A reaches the same level as the top of tube G, this tube becomes a syphon and seeks to draw off the oil in chamber A. However, the upper part of chamber A communicates with the carburetor intake through tube F, and therefore is under vacuum, and as long as this vacuum exists the syphon cannot become operative.

When float D rises, the valve I at the top of tube E is opened and air from the crankcase rushes in through E to raise the pressure in chamber A to atmospheric. At this point the syphon in tube G becomes operative and draws off all of the oil in chamber A. With the oil level again lowered, float D lowers to close valve I, and the operation of distillation in chamber A is repeated.

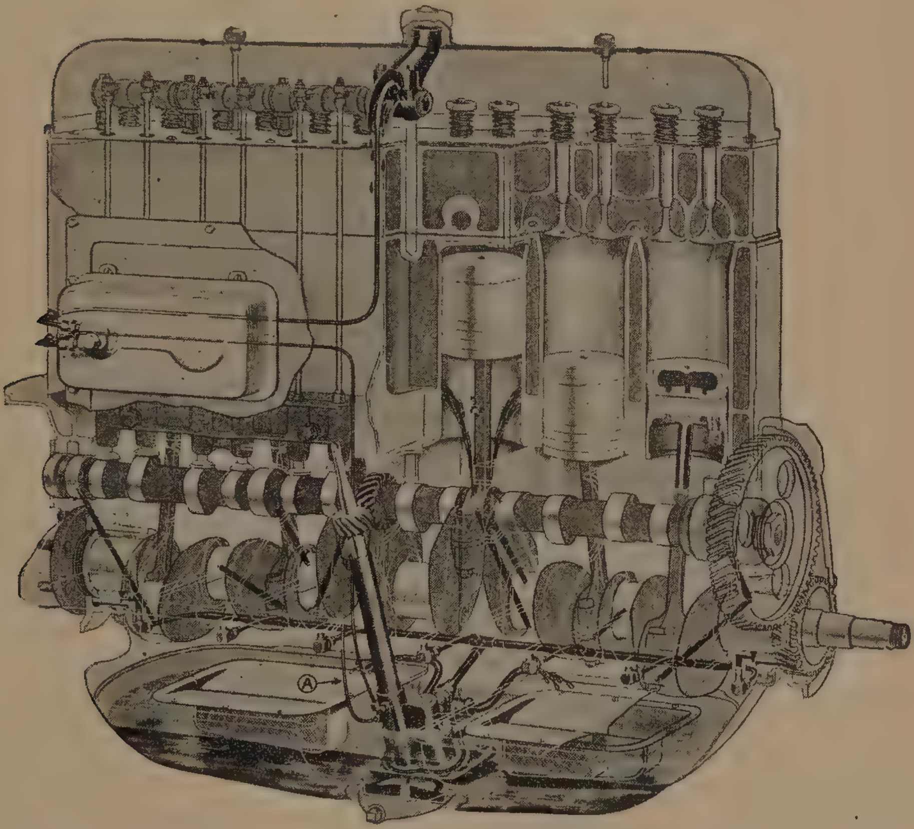


Fig. 293a.—Phantom View of Nash Engine-Lubrication System.

CHAPTER XVI

Starting Crank, Manifolds and Muffler

Practically all passenger cars and most of the smaller trucks are now provided with electric starters, and a hand starting crank is carried for emergencies only. Large trucks, however, still have their engines started by hand. Wherever hand cranking is used regularly the hand crank remains in position at the front of the car, whereas when it is used only in emergencies it is carried in the tool box and the opening at the front of the car through which it is inserted is closed by a cap.

Starting Crank Ratchets—The ratchets of starting cranks are made in many different forms, several of which are illustrated in Fig. 294. That shown at B is formed on the shaft of the starting crank itself, which is drilled out for the purpose, and has five end ratchet teeth milled on it. Frequently the engaging faces of these teeth are in a plane passing through the axis of the shaft. It is a much better plan, however, to make them at an angle of about 10° with that axis, as in designs A, B, and D, so that the teeth of the two ratchets will interlock, for even though these ratchets are hardened, their teeth will wear in time, and when the points of the teeth have worn back the ratchet is liable to disengage while the operator is pulling hard on the starting crank, which is extremely annoying, if not dangerous. It may here be pointed out that practically all engines turn right-handedly when looked at from in front of the car, or the starting end.

Starting crank ratchets for four-cylinder engines should preferably have two teeth, spaced 180°, so that the starting crank handle will always be in the same position when the crankshaft passes the dead center. For the same reason the ratchet for six-cylinder starting cranks should have three ratchet teeth, spaced 120°, as shown at D. In some cases the ratchet on the end of the crankshaft is replaced by a pin passing through same, with projecting ends with which the teeth on the starting crank ratchet engage, the teeth then being made of special form, as shown at C, Fig. 294. Generally the ratchet on the forward end of the crankshaft is keyed or pinned to the shaft, but sometimes, in order to save space in the longitudinal direction, it is formed integral with the crankshaft pinion (see A, Fig. 294). Another plan that is much followed consists in making this ratchet in the form of a nut which holds in position other parts on the forward end of the crankshaft, such as the timing gear pinion and fan drive pulley.

Angular Relation of Starting Crank to Crankshaft—In order to render the starting of the engine as easy and safe as possible the ratchets on the starting crank and the crankshaft must occupy certain definite angular positions. The resistance which the operator must overcome in turning the engine over by hand is due partly to the friction of the pistons in the cylinders and partly to the compression in one cylinder, the latter being generally the most important factor. The compression reaches its maximum value when the piston is at the top end of the stroke, but the crank effort necessary at that moment is zero, owing to the toggle effect of the crank and connecting rod. The crank effort required at various angles of the crankshaft during the compression stroke is shown by the diagram, Fig. 295, from which it will be seen that this effort reaches its maximum when the crank is about 25° ahead of dead center.

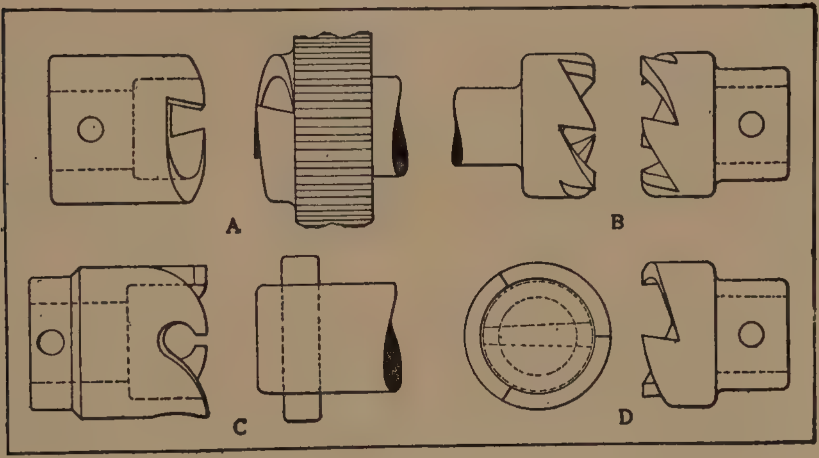


FIG 294.—FORMS OF STARTING CRANK RATCHET

The usual method of starting the engine consists in pulling up on the starting crank, and the operator undoubtedly applies his strength to the best advantage when the starting crank is in the horizontal position or half-way up. Hence the ratchets on the starting crank and the crankshaft should be so arranged that the starting crank is half-way up, or 90° from the straight-up position, when one set of cranks has to travel 25° more till it reaches the top dead center position. A four-cylinder crankshaft usually stops in such a position that all of its pistons are about mid-stroke, hence the crankshaft must be turned through a quarter-circle to bring it to the dead center or firing position, and since, when the crankshaft is in the dead center position the starting crank will be $90 - 25 = 65°$ from the straight-up position, it will generally be possible to pick up the crankshaft at $65 + 90 = 155°$ from the straight-up or 25° from the straight-down position (Fig. 296).

The turning moment which it is necessary to impress upon the starting crank is proportional to the square of the bore and to the length of the stroke. It is, of course, impossible to make the length of the starting crank proportional to the turning moment required, as that would give exceedingly small cranks for small engines and entirely too large cranks for large ones. The following formula gives suitable values for the length of the crank:

$$L_c = d\sqrt{l}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (92)$$

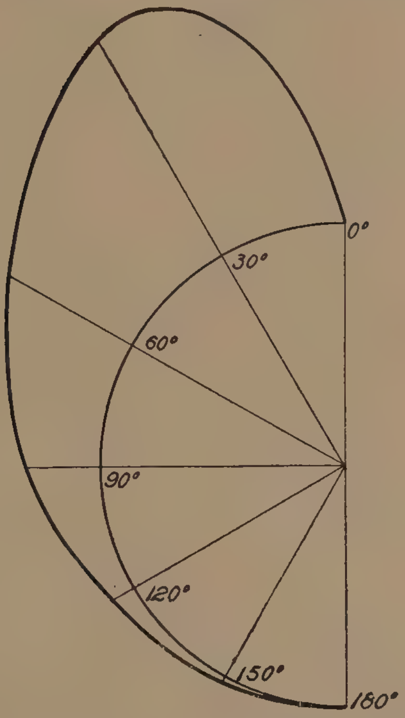
The necessary cross-section of the crankarm may be calculated on the basis of 12,000 lbs. per sq. in. stress in the metal when the operator applies an effort of 100 lbs. to the crank handle. Supposing the starting crank to be 10 in. long, the bending moment is

$$10 \times 100 = 1000 \text{ lb.-in.}$$

The section of the crank must, therefore, have a resisting moment of 1000 lb.-in. when subjected to a fiber stress of 12,000 lbs. per sq. in. In the case of a 4 × 5 in. engine, making the thickness or smallest dimension of the arm ½ in., the width or largest dimension near the hub figures out to about 1 in., if the arm is made of solid rectangular cross-section.

Starting cranks for passenger cars, being for occasional use only, are now generally of very simple design, bent to shape from round section stock, of uniform section throughout, and without a revolving grip. For constant service the starting handle should be about 1 in. in diameter and not less than 5½ in. long. It should be provided with a brass tubular or hardwood grip which turns freely on the pin or stud, so as not to injure the operator's hands. The handle is sometimes drilled out for lightness and is generally expanded into the crankarm.

Locking Starting Crank—The starting crank must not be allowed to hang down when the car is in motion, as its grip is generally considerably lower than the front axle, and is therefore likely to be struck by obstructions on the road.



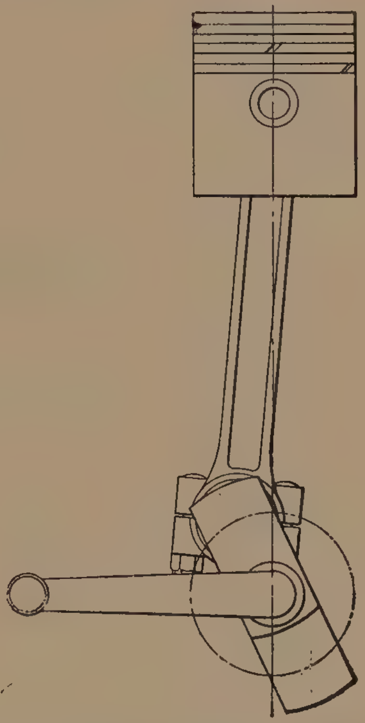


Fig. 295 (left).—Effort Required on Starting Crank at Different Crankshaft Angles.

Fig. 296 (right).—Relative Angular Position of Crankshaft and Starting Crank.

On motor trucks it is not desirable that the handle of the starting crank extend forward when not in use. For convenience in starting, the handle must project ahead of the bumper with which most trucks are fitted, and if it does this while not in use it is apt to be damaged by collision with some object. To prevent such damage the shaft of the crank on trucks is generally provided with a hinge joint, so that after the engine has been started the crank can be swung back and held in a clip secured to the front cross member of the frame. Figure 297 shows one construction serving this purpose. Instead of the shaft of the crank being hinged, it

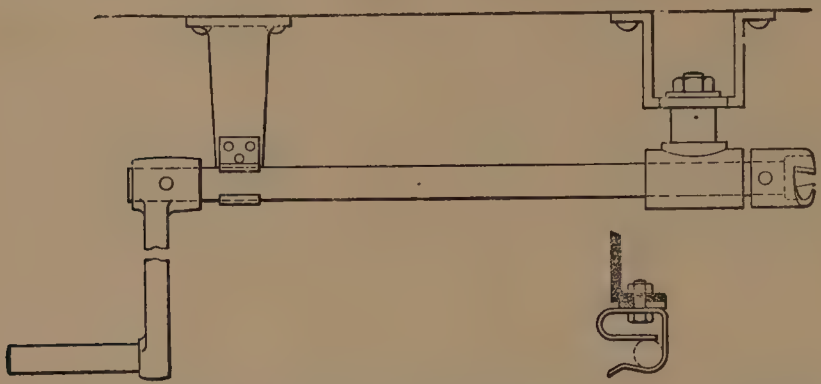


FIG. 297.—STARTING CRANK SUPPORTING CLIP AND SWINGING BRACKET.

is supported in a rotatable bearing. The sub-figure shows a simple form of clip for this purpose.

Figure 298 shows the variation of the starting and low speed running resistance of automobile engines as a function of the total cylinder displacement in cubic inches. This diagram, which is taken from a paper read by C. E. Wilson before the Society of Automotive Engineers, is of value in the design of self-starters.

Manifolds—The design of manifolds has undergone important changes during the past 15 years, as a result of the

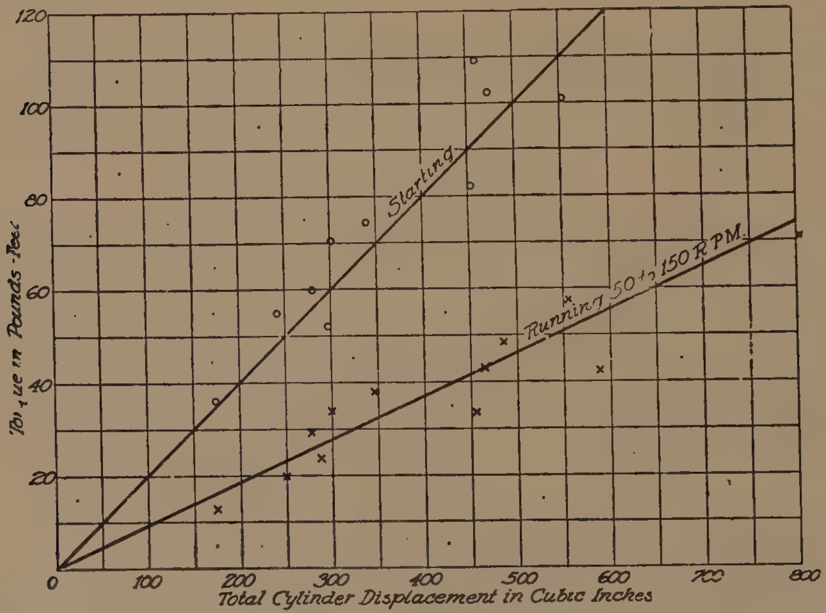


FIG. 298.—DEPENDENCE OF STARTING TORQUE ON PISTON DISPLACEMENT.

change in the character of the fuel marketed. Previous to 1910 all engines had inlet and exhaust manifolds that were entirely independent of each other, and the design shown in Fig. 299 is typical of the manifolds then used on four-cylinder engines. A few of the lower priced cars of to-day have manifolds of substantially the same type. When it became desirable to supply heat from the exhaust to the induction system in order to properly vaporize the fuel, air heaters or air stoves were provided, which consisted of a muff surrounding a section of the exhaust manifold or exhaust pipe, through which the carburetor air was drawn, the connection from the stove to the carburetor air inlet usually being by a flexible metallic tube.

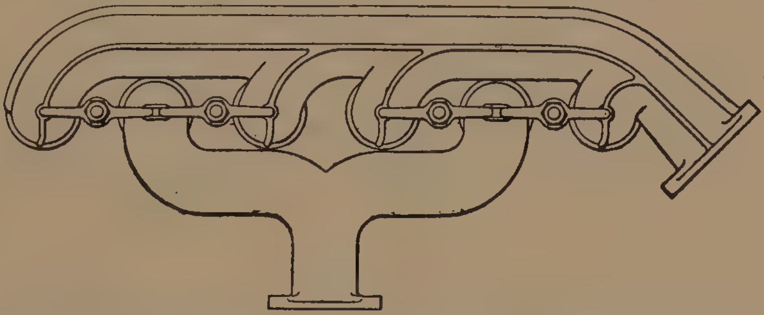
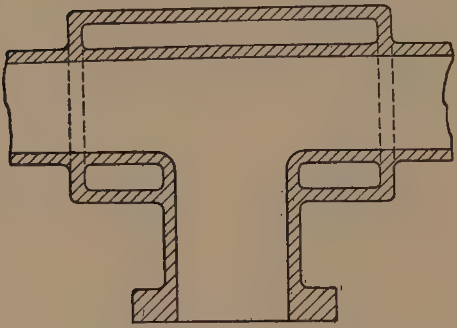


FIG. 299.—INDEPENDENT INLET AND EXHAUST MANIFOLDS.

Later engineers came to the conclusion that this was not the best way of supplying the heat. The heat, of course, is wanted for the fuel, which has to be vaporized, and if the air is used as a medium of heat transfer, it will be heated more than the fuel. Now, heating the air expands it, and the hotter the air, the less the quantity which is drawn into the cylinder, and the less the power which can be obtained from the engine. It is therefore a better plan to supply the heat to the fuel directly, by surrounding that part of the inlet manifold at which any unvaporized fuel collects, with a jacket through which either all or a part of the exhaust gases can be passed. It has been found that where there is a T or right-angled turn in the manifold the unvaporized fuel collects at the inner corner, and that is the best place to apply the heat. At this point the inlet is generally jacketed as shown in Fig. 300. When this method of applying exhaust heat to the vaporization of the fuel first came into use, the inlet and exhaust manifold were usually made with a common wall at a point where the incoming mixture was compelled to make a sharp turn and where the unvaporized fuel particles separated out by reason of their inertia. This constituted what was known as a hot spot vaporizing system, an example of which is illustrated in Fig. 301.

Charge Distribution—Inlet manifolds must be so designed as to ensure as nearly as possible equal quantitative and equal qualitative distribution. Equal quantitative distribution means that all cylinders of the engine receive the same amount of charge, while equal qualitative distribution means that the ratio of fuel to air in the charge is the same in all cylinders. To ensure equal quantitative distribution, the resistance to gaseous flow from the carburetor outlet to all cylinders must be substantially the same. Naturally, the resistance to the flow is somewhat greater in the case of the end cylinders. The effect of this can be minimized by making the resistance of the manifold small as compared with the resistance of the valve opening. However, the area of the inlet manifold section must not be made too large, as that makes it impossible to keep the only partly vaporized fuel in suspension at low speeds, and interferes with idling. In engines with six cylinders or less the cross-section of the inlet pipe is generally made equal to that of the valve port. With eight-cylinder engines, if a single carburetor is used, the inlet manifold section at the carburetor must be about 30 per cent larger than the inlet valve port area. The cross-section of the exhaust manifold is usually made about 50 per cent larger than that of the inlet manifold.



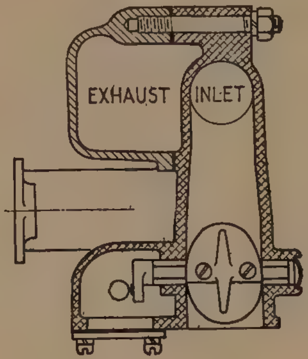


Fig. 300 (left).—Inlet Manifold Jacketed at T.
Fig. 301 (right).—Showing Principle of Hot-Spot.

That uniform distribution is very essential to maximum power development and is hard to secure with multi-cylinder engines is indicated by the fact that on eight-cylinder racing-car engines a separate carburetor is sometimes used for each

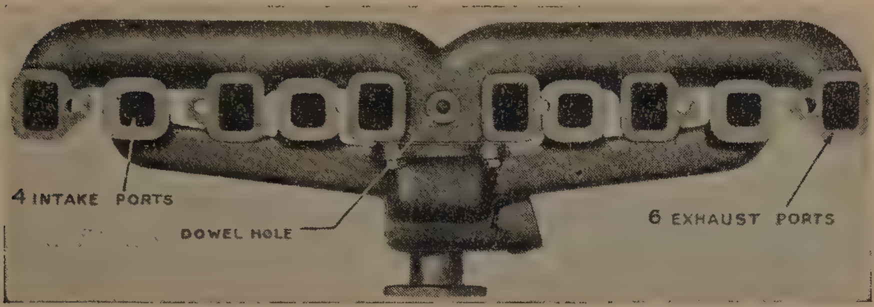


FIG. 302.—CLEVELAND MANIFOLD ASSEMBLY.

cylinder. This, of course, completely eliminates the distribution problem. The inlet pipes can then be made as large as required for maximum power, and each carburetor can be adjusted to deliver the best possible mixture to its particular cylinder. European manufacturers of six- and eight-cylinder passenger cars often fit their engines with two carburetors, but dual carburetors are unfamiliar in this country.

Heating by Exhaust Gases—In order to ensure satisfactory distribution in a multiple-cylinder engine, it is absolutely necessary to supply heat to the charge before it enters the distributor branches. Yet, this heat must not be applied too near the carburetor, as in that case some of the more volatile fractions of the fuel in the carburetor may vaporize and interfere with the metering.

Either all or a part of the exhaust gas may be circulated though the inlet manifold jacket, or means may be provided whereby all of the gas can be passed through when starting

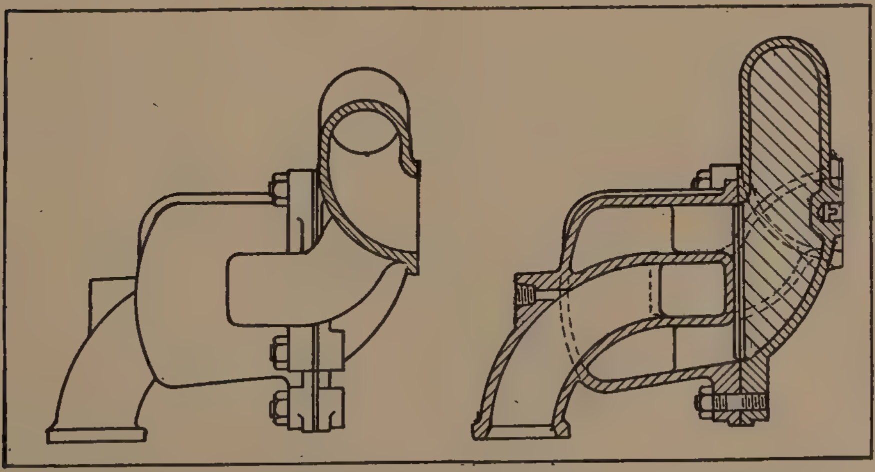


FIG. 302-A.—TWO SECTIONS THROUGH CLEVELAND MANIFOLD.

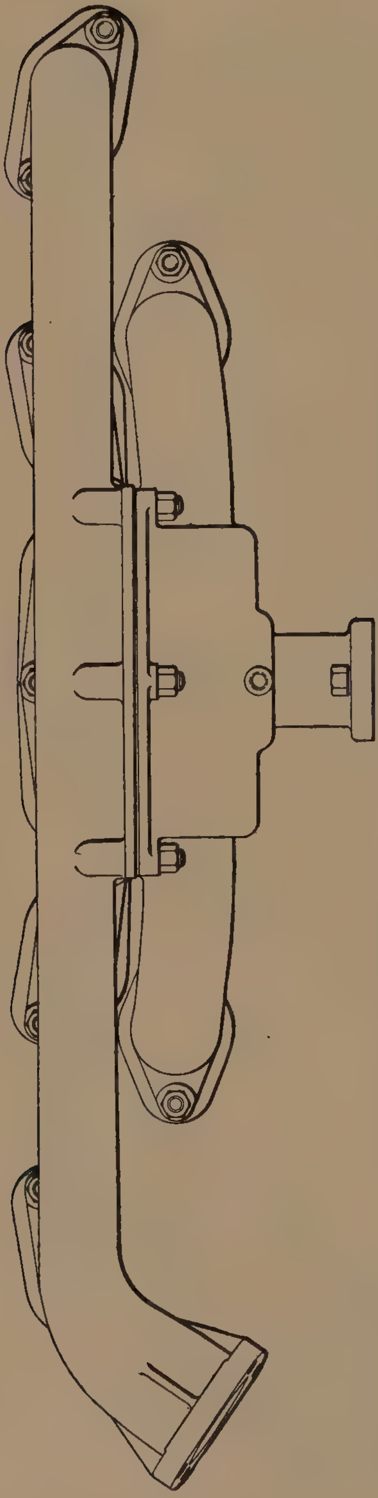


FIG. 303.—MANIFOLD OF LOCOMOBILE SIX-CYLINDER ENGINE.

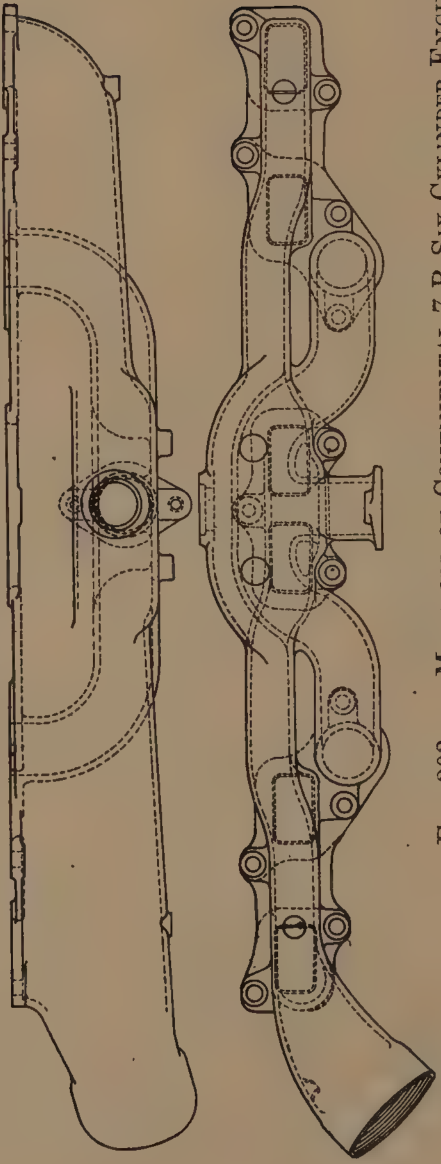


FIG. 303-A.—MANIFOLD OF CONTINENTAL 7-R SIX-CYLINDER ENGINE.

and idling, and only a portion or none at all when running under full power. If all of the exhaust is to pass through the jacket the branches of the manifold must converge toward the exhaust jacket on the inlet manifold at the middle of the engine and the exhaust pipe lead from it. On the other hand, if the down-take of the exhaust pipe is at the rear end of the engine, only the exhaust from the forward cylinders will be carried through the jacket. The former scheme is used on the Cleveland engine, the manifold of which is illustrated in Fig. 302. Two transverse sections of the manifold are shown, from which the provision for heating the incoming charge may be seen.

The Marvel Carburetor Co., has worked out a system of heat control by means of two valves which can be set from the instrument board, by which all or any portion of the exhaust can be passed through the jacket on the inlet pipe. This is used on several popular makes of car.

A typical arrangement of a manifold on a six-cylinder passenger-car engine (Locomobile) is shown in Fig. 303. This engine has six exhaust ports on an upper level and two inlet ports on a lower level. This, of course, means that three inlet valves communicate with each other through a cored passage in the cylinder block. Inlet and exhaust manifolds are cast separately and bolted together at the middle, the former being located below the latter. At its middle portion the inlet manifold is cast with a jacket which is open to the exhaust manifold. At first a baffle was provided in the exhaust manifold to deflect the stream of exhaust gases from the forward cylinders down into the jacket space, but this was later found to be unnecessary and was omitted. A very similar manifold is used on the Peerless six-cylinder engine, which however, is made in a single casting.

Exhaust Jacket Area Required—A Committee of the S. A. E. which some years ago investigated the best methods of dealing with the fuel then on the market, recommended that in hot-spot manifolds the area of the exhaust-heated surface should be 1 sq. in. per 30 cu. in. of piston displacement, provided the surface were near the exhaust ports; if it were 12 in. or more from the nearest exhaust port, then 1 sq. in. of exhaust heated surface should be provided per 15 cu. in. displacement.

It has been observed (with glass manifolds) that the unvaporized portions of the fuel always collect at the inner corners of turns, contrary to what might be expected, and a number of designers therefore make provision to catch this fuel and subject it to an intense heating effect, whereby it is vaporized and caused to mix with the charge flowing into the cylinders. This is accomplished by providing grooves or pockets on the inner side of the bend and exhaust-jacketing the wall at this point. The arrangement used on the Marmon, Fig. 304, is typical.

Experimental investigations have shown that (with fuel of 450° F. end point), some of the fuel enters the cylinder in the liquid state under practically all working conditions, and heat from the piston head and cylinder walls as well as the heat of compression must be depended upon to vaporize it. In the design of the inlet manifold the main aim should be to keep the air and fuel thoroughly mixed, so that all cylinders will get equal charges of fuel. The liquid fuel tends to adhere to the walls, and the air undoubtedly flows fastest in the center of the passage. Therefore, some designers have recommended to provide a throat with a sharp edge at the junction of the horizontal and vertical sections, or to cast baffles on the inside of the pipe which tend to turn any fuel collecting on the walls, back into the middle of the air stream. Some advantage has been claimed also for square as compared with round-section manifolds, and for sharp as compared with easy bends.

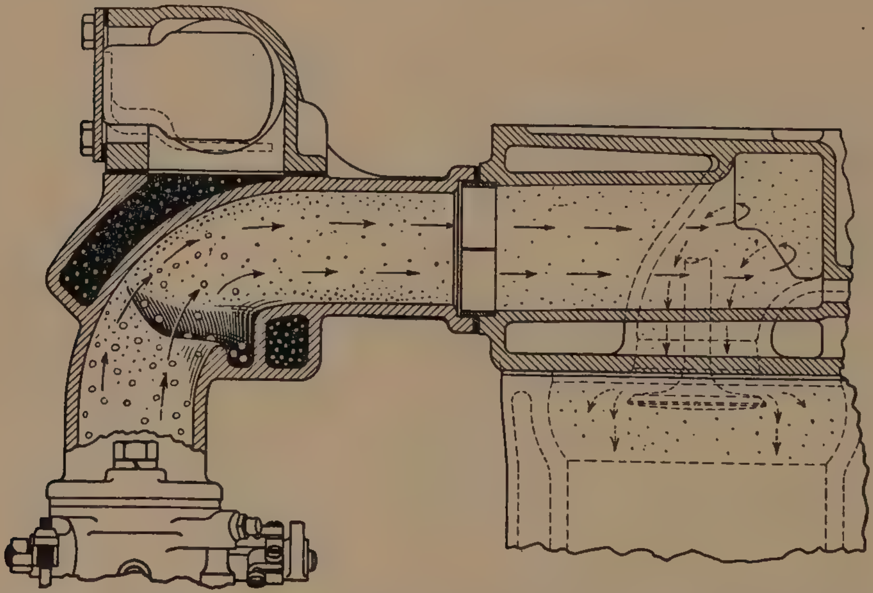


Fig. 304.—Section through Marmon Inlet Manifold Showing Trough at Knee Designed to Catch Unvaporized Fuel and Boil it Off.

Flanges for inlet manifolds have been standardized by the S. A. E. (See Appendix).

In quite a number of passenger cars the exhaust pipe is carried down at the forward end, rather than at the rear, which latter would in one respect be the most logical place, because it has to connect to the muffler which is always carried on the frame under the body, that is, to the rear of the engine. The reason for carrying it down in front is to get it further away from the body, so not so much heat will be transmitted to the body, to the annoyance of the driver in hot weather.

Effect of Firing Order—The explosions in a four-cylinder engine may be timed in either of two orders, viz.:

1—2—4—3, 1—3—4—2.

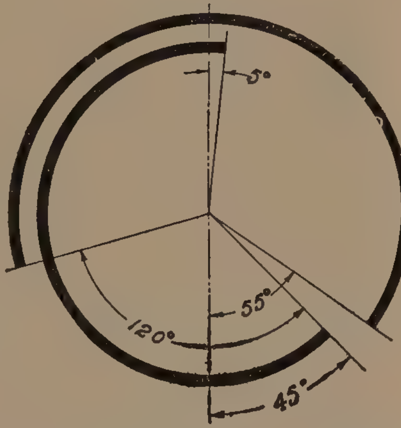


Fig. 305.—Showing Overlapping of Exhaust Periods in Six-Cylinder Engine.

Neither of these firing orders has any appreciable advantage over the other.

The firing order is of more importance in a six-cylinder engine. We found that the exhaust valves remain open for about 230° of crank motion. Now, in a six-cylinder engine the exhaust valves begin to open at intervals of 120°, so two exhaust valves are open simultaneously practically all the time. This overlapping of the exhaust periods is shown graphically in Fig. 305. Suppose that the exhaust valve opens with 45° lead and closes with 5° lag; then, when the exhaust valve of one cylinder begins to open the exhaust valve of the preceding cylinder is still fully open, and remains open during another 110° of crank motion, so that if the two cylinders are adjacent to each other and connected to the same exhaust manifold, and especially if the exhaust ports of the two cylinders communicate directly inside the cylinder, exhaust gases must "blow over" from the cylinder just beginning to exhaust into the one completing its exhaust stroke, thus interfering with the evacuation of the latter. This effect is brought out very clearly in Fig. 306, which shows an indicator diagram of a six-cylinder engine.

This difficulty can be partly overcome by so arranging the firing order that the two cylinders in one pair never fire in succession. For instance, the following three firing orders may be used:

1—4—2—6—3—5,
1—5—3—6—2—4,
1—3—2—6—4—5.

With the first two of the above firing orders explosions never follow one another in the three cylinders at one end.

The surest way to prevent interference of the exhausts from succeeding cylinders is to provide two exhaust manifolds, one for the three forward and the other for the three rear

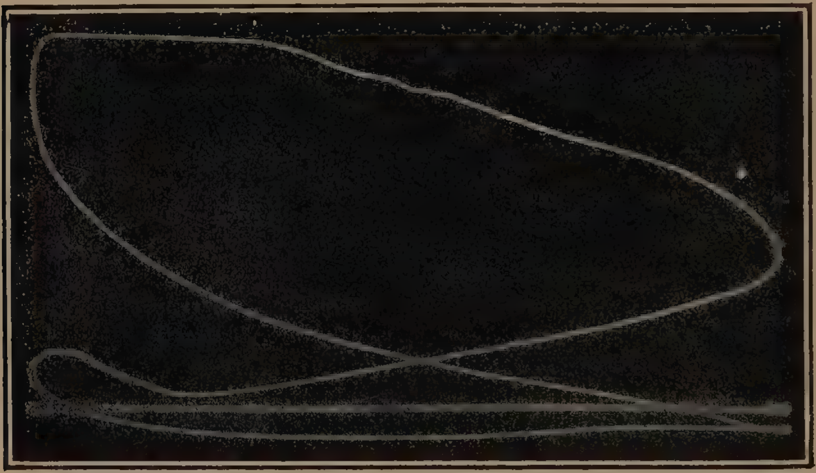


Fig. 306.—Showing Effect of "Blow-Over" from One Cylinder into Another.

cylinders, connecting to a double exhaust pipe leading to the muffler.

Firing Orders of V Engines—In V engines the valves of oppositely located cylinders are operated by a common cam, generally through the intermediary of bell cranks, or the camshaft is provided with one cam for each valve to be lifted. In the former case the number of firing orders is limited. For instance, in an eight-cylinder 90° V engine, if the crankshaft turns right-handedly (looked at from the front), any left-hand cylinder (looked at from the rear) must fire either 90° or 450° of crankshaft motion after the corresponding right-hand cylinder. These motions correspond to 45° and 225° of camshaft motion. Now it would be impossible to get the two cam followers on the cam so close together that one of the valves would begin to lift 45° of cam motion after the other, owing to interference between followers, hence the left-hand valve must begin to lift 225° of cam motion after the right-hand valve. As an explosion occurs every 45° of cam motion, this means that the left-hand valve of a given pair of cylinders will be fifth to lift after the corresponding right-hand valve. Combining this order with the two possible firing orders for each set of four cylinders we get the following two possible firing orders for an eight-cylinder engine in which a single cam operates two valves:

1 *R*—4 *L*—2 *R*—3 *L*—4 *R*—1 *L*—3 *R*—2 *L*
1 *R*—4 *L*—3 *R*—2 *L*—4 *R*—1 *L*—2 *R*—3 *L*

The latter of the two firing orders is generally used, probably because it happened to be adopted by one of the pioneers in the V type engine field.

With eight-cylinder vertical engines the interval between explosions also is 90°. An eight-cylinder vertical engine may be considered as made up of two four-cylinder engines of which one is located at the middle and half of the other at each end. There are two possible firing orders for each "four," and since each firing order in one of the fours can be combined with each firing order in the other four, this makes four possible firing orders. However, the number of possible firing orders is doubled once more because of the fact that No. 1 cylinder of one "four" may be followed by either No. 1 or No. 4 of the other "four," which would be numbered 3 and 6 in the eight-cylinder engine. Thus with one and the same crankshaft design there are eight possible firing orders. Four of these are as follows:

1—3—2—4—8—6—7—5
1—3—7—4—8—6—2—5
1—6—2—4—8—3—7—5
1—6—7—4—8—3—2—5

The other four are obtained by interchanging Nos. 4 and 5, leaving everything else the same.

In an eight-cylinder engine the overlapping of exhaust periods is even greater than in the "six." In a V engine it is customary to have entirely separate exhaust manifolds and mufflers for the two sets of cylinders, so this overlapping causes no trouble. In a vertical eight the exhaust manifold may be made in halves, one for the forward and the other for the rear cylinders, with take-downs at front and rear respectively, or else the take-down may be at the middle and the two pipes joined only at some distance from the engine ports, as in the Panhard manifold illustrated in Fig. 307.

Muffler Design—Automobiles must operate substantially noiselessly, and the gases from the exhaust must be passed through a muffler before they are discharged to the atmosphere. In the muffler the gases must be allowed to expand gradually and to cool, thereby reducing the pressure, which is the cause of the noise when they are discharged into the atmosphere directly. While it is an easy matter to so obstruct the passage from the cylinder to the atmosphere that the gases will be discharged without disagreeable noise, they must escape with comparative freedom, so that there will be no back pressure from them on the piston during the exhaust stroke, and this complicates the problem of muffler design.

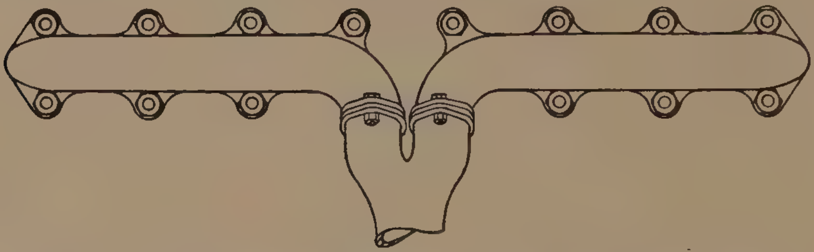


Fig. 307.—Bifurcated Exhaust Manifold (Panhard).

The muffler is generally carried under the rear part of the frame of the car, as far away from the engine as possible, so the engine will exhaust into a pipe whose capacity is two to four times that of a single cylinder. This gives the hot gases a chance to expand and cool. It is, however, advisable to provide additional expansion volume in the muffler. The combined volume of the exhaust pipe and that compartment of the muffler which the gases enter first should not be less than six times the piston displacement of a single cylinder. This expansion chamber should have as large a surface as possible in contact with the atmosphere, so that heat may be rapidly abstracted from the gases. From the expansion chamber the gases are led through another chamber or series of chambers communicating by means of fine and sometimes tortuous passages, before they are finally discharged to the atmosphere. Sometimes these muffling chambers are placed concentric with the expansion chamber and sometimes they are placed adjacent to it.

The muffler should preferably be so arranged that it can be taken apart for cleaning, since lubricating oil and solid carbon particles held in suspension by the exhaust gases will be deposited on its walls, clogging the fine passages and thus increasing the back pressure. It must, however, be of sufficiently strong construction to withstand the pressure of an explosion of a gasoline-air mixture at atmospheric pressure, since such explosions in the muffler are of common occurrence. The walls must be so designed that they will not be set into harmonious vibration by the succeeding exhausts, and especially should they not be of any material with sonorous properties. A covering of sheet asbestos over the muffler will prevent ringing, but is undesirable on account of its heat-insulating properties. The outer cylindrical shell should preferably be made of very thin and soft sheet iron formed into a roll of several layers, held together by rivets. Mufflers are made of a total capacity of from 12 to 20 times the capacity of one cylinder.

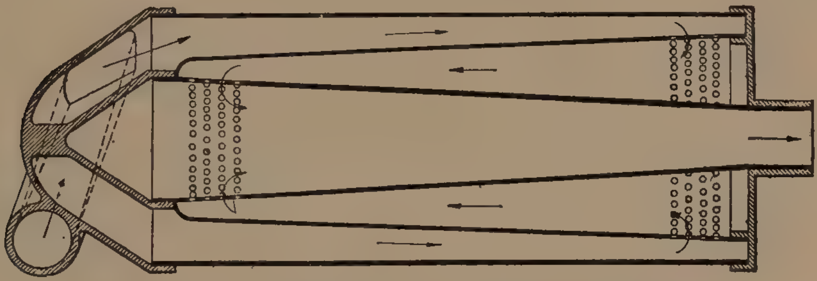


Fig. 308.—Concentric-Tube Type of Muffler.

Concentric Tube Types—A type of muffler that is known to give very satisfactory results when properly proportioned is shown in Fig. 308. It consists of two cast-iron heads which are connected by three concentric sheet metal tubes, the two inner ones of which are here shown conical, but may, of course, also be cylindrical. The gas enters the outer one of the three concentric chambers through a spiral passage cast on one of the heads. They expand in this chamber and then pass through fine perforations at the opposite end in the partition wall into the intermediate chamber, and thence through similar perforations at the near end of the innermost partition wall into the central chamber, from which they escape at the further end. This muffler is a development of an earlier type, also of concentric cylindrical chamber form, in which the gases entered the innermost chamber and were discharged from the outermost chamber. It has the advantage over the latter type that it is easier to provide a comparatively large initial expansion chamber, and that the walls with which the hot gases come in contact are exposed to the atmosphere, hence the gases are cooled more rapidly. The swirl given to the gases by the spiral inlet passage brings the newly entering charge immediately in contact with the entire outer wall of the muffler and prevents any direct impact between the column of charge and a wall of the muffler.

The method of keeping the parts in place is not illustrated, but preferably rods would be used, extending lengthwise through the muffler, with nuts on their ends.

Another form of muffler is shown in Fig. 309. This consists of a considerable number of flat cylindrical chambers of pressed steel with one head integral and the edge at the open end expanded so as to fit over the closed end of the adjacent section. The heads are provided with openings through which adjacent chambers communicate, these openings being alternately at the center and near the circumference, thus compelling the gases to follow a tortuous course through the muffler. The heads are made of cast iron and the gases enter and leave at the centers of the heads. The first and last partition walls differ from the others, the first having openings both at the center and near the circumference to facilitate rapid expansion, and the last has fine perforations only, so as to break the discharge up into a large number of fine streams.

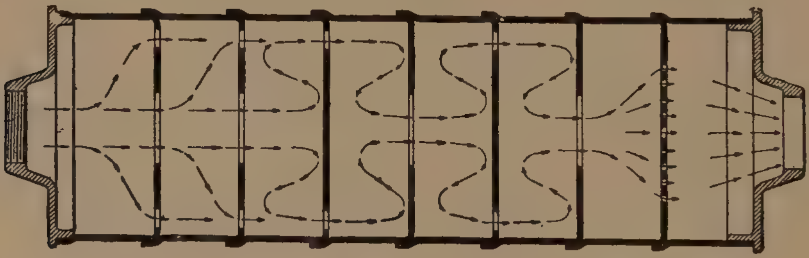


Fig. 309.—Pressed-Steel-Cup Type of Muffler.

Figure 310 illustrates the muffler used on the Cadillac car. It is made entirely of sheet metal—pressings and deep drawings. The gases enter the expansion chamber through a nozzle with perforated wall. The casing is in four parts, two cylindrical and two bell-shaped. The two cylindrical parts telescope together, and the joint is welded. All three are flanged. Five pressed steel baffles are located inside the case. These also are flanged and are clamped with their flanges between the flanges of the case members. The two bell-shaped casing members are identical in shape, which simplifies manufacture. This muffler, of course, can be readily taken apart, as its members are held together by bolts. The shape of all members is such as to give great resistance to gaseous pressure, although they are comparatively light.

Figure 311 shows the muffler used on the Locomobile. It consists of a 6-in. thin-walled tube with sheet steel headers pressed each with a circular corrugation where they are joined to the tube by welding. Two 2¼-in. tubes with perforated walls extend through the larger tube, abutting against one head and extending through the other, into which they are welded.

Where the tubes enter the muffler asbestos cord packings are provided, and the opposite ends of the tubes are held rigidly in place by discs bolted to the headers, of the right size to just enter the tubes.

Back Pressure of Mufflers—Several competitions with the object of improving the design of mufflers were held by the Automobile Club of France during the early years of the industry. In one of these the tests developed the interesting fact that a length of pipe connected to the exhaust manifold of the engine may increase the power output. During the first part of the trial all tests were made with the mufflers connected directly to the exhaust manifolds. It was then decided to place them at some distance from the motor, to reproduce the conditions on an automobile, and the connection was made by a pipe with three bends in it, 6 feet long and of 2 inches internal diameter, substantially the same as the manifold diameter. The engineer in charge of the tests anticipated that a certain correction would have to be made for the loss in this pipe, and proceeded to determine this loss. He was greatly surprised to find by repeated experiments that this pipe increased the power, and accurate tests showed the increase to be 1.4 per cent. The explanation offered for this phenomenon is that when the gases pass through the 6 foot pipe their temperature is reduced to quite an extent and they

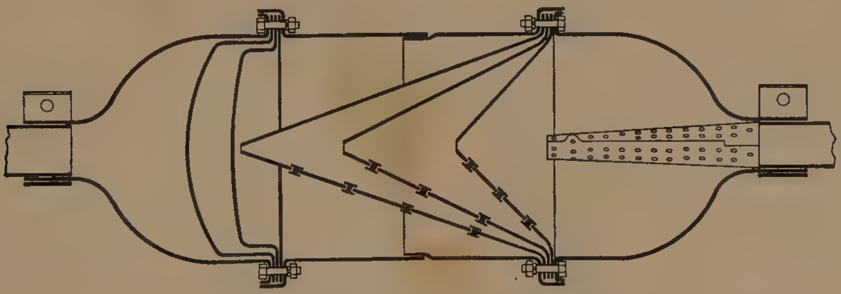


FIG. 310.—CADILLAC MUFFLER.

issue from the mouth of the pipe at a considerably lower speed than if discharged directly from the exhaust port, hence the resistance to their entrance into the atmosphere is less, which, of course, means less back pressure. The most silent mufflers in this competition reduced the power of the engines 10 to 11 per cent.

Tests conducted in the automobile laboratory of the University of Michigan in 1914, showed that while some mufflers at full engine load absorb 14 to 17 per cent of the engine power, the average percentage of the power absorbed throughout the whole range of engine output is about 3.

On the Chrysler "80" a high-speed car on which low back pressures of the exhaust system are essential, the back pressures at different car speeds are as follows: 20 m.p.h., 0.05 lb. gauge; 30 m.p.h., 0.20; 40 m.p.h., 0.37; 50 m.p.h., 1.00; 60 m.p.h., 2.00; 70 m.p.h., 4.

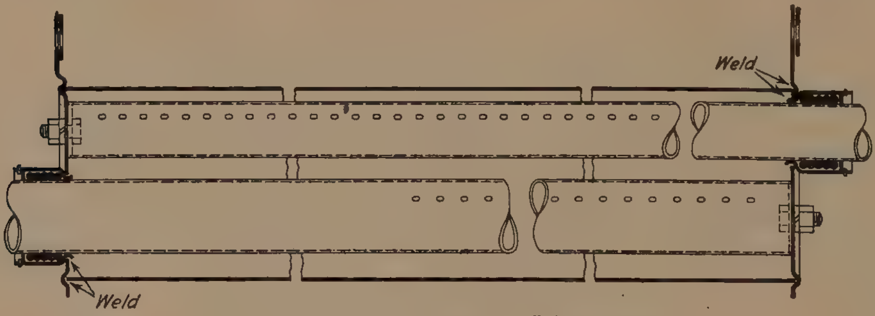


FIG. 311.—LOCOMOBILE MUFFLER.

The final outlet from the muffler is generally through a pipe of considerably less cross-section than the exhaust pipe, which tail pipe is sometimes tapered or reduced in section away from the muffler.

The chief requirements in muffler design may be summarized as follows: Use non-sonorous materials, such as soft sheet iron and cast iron. Provide an expansion chamber of large size as compared with the volume of the individual cylinder. Keep the hot gases during the first part of their passage through the muffler in contact with surfaces exposed to the atmosphere so as to reduce their temperature and pressure as quickly as possible. Gradually decrease the cross-sectional area of the passages so that the exhaust will issue from the outlet pipe in a practically continuous stream. Make the muffler of sheet steel pressings as far as possible, so as to reduce the cost of manufacture and use bolted joints for the casing so the muffler can be taken apart for cleaning.

CHAPTER XVII

The Flywheel

As was explained in Chapter III, in a four-cycle engine there is only one explosion or power stroke in every four strokes of the piston. Consequently, in a single-cylinder engine a turning moment is impressed upon the crankshaft during only one stroke out of four, and there are three idle strokes between power strokes.

In order to have a practically usable engine, the turning moment or torque must be substantially uniform throughout the cycle, and this makes it necessary to store up some of the mechanical energy developed during the power stroke and draw upon this store of energy during the three idle strokes, so that the rate of doing external work is practically constant. The flywheel forms this means of storing energy.

Variation of Crank Moment—Fig. 312 shows a crank moment diagram for a single-cylinder $3\frac{1}{2} \times 5$-in. engine based upon a diagram showing an explosion pressure of 340 lbs. per sq. in. and on a 10-in. connecting rod. The diagram shows the crank moment at low speeds, and the effect of the inertia of the reciprocating parts on the crank moment is neglected, because the flywheel function is most important when the engine runs at low speed, when the inertia effects of the reciprocating parts are small.

The cylinder develops approximately 415 ft.-lbs. of energy per explosion, net; that is, the indicated energy of the explosion stroke is 474 ft.-lbs. and that of the compression stroke 59 ft.-lbs., leaving 415 ft.-lbs. of available energy. Now consider a crank with an arm 1 ft. in length. The above amount of energy—415 ft.-lbs.—is expended during two revolutions of the engine crankshaft, and during these two revolutions the outer end of the 1-ft. crank travels

$$2 \times 2 \times 3.14 = 12.56 \text{ ft.}$$

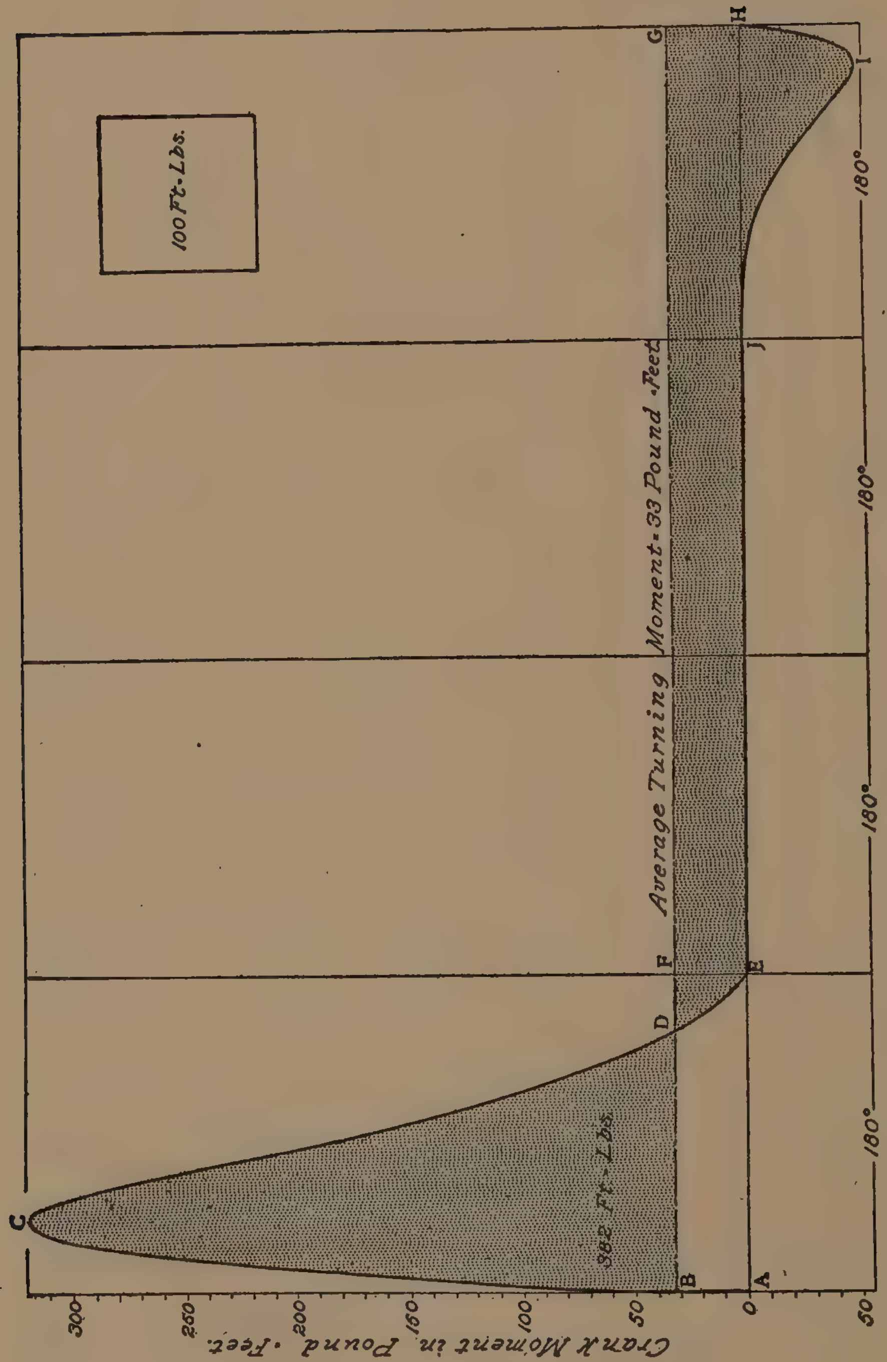


FIG. 312.—LOW-SPEED CRANK-MOMENT DIAGRAM FOR SINGLE CYLINDER $3\frac{1}{2}$ BY 5-IN. ENGINE.

The force acting at the outer end of the 1-ft. crankarm is therefore

$$\frac{415}{12.56} = 33 \text{ pounds, (approx.)}$$

Consequently, the average turning effort or torque is 33 lb.-ft. We may consider the resisting moment of the load on the engine to be constant, and its value, then, is 33 lb.-ft.

Referring now to Fig. 312, it will be seen that while the resisting moment or torque load is only 33 lb.-ft., the torque impressed upon the engine crankshaft during the power stroke rises as high as 320 lb.-ft. The excess torque goes to accelerate the flywheel and to store up energy in it. In the figure, the area A B C D E represents the total energy of one explosion = 474 ft.-lbs. The area B C D represents excess energy developed during the first four-fifths of the explosion stroke, which is stored in the flywheel. This area represents 382 ft.-lbs. The energy stored up in the flywheel during the power stroke is given out again during the three so-called idle strokes, doing the external work during these three strokes, represented by the rectangle E F G H, and the work of compression represented by the area H I J. In reality the flywheel begins to supply some of the energy necessary to perform the outside work at the point D, before the end of the power stroke. The area D F G H I J E, representing the total energy given out by the flywheel during one cycle, is exactly equal to the area B C D, representing the energy absorbed by the flywheel during one power stroke.

Constructions similar to Fig. 312 enable us to determine the relative amount of flywheel capacity theoretically required for engines with different numbers of cylinders. In Fig. 313 is shown a crank moment diagram for a double-cylinder opposed 3½ × 5-in. engine, other conditions being the same as in Fig. 312. The crank moment during only one revolution is shown, as the diagram repeats itself every revolution. Measurement shows that in this case the flywheel has to absorb 302 ft.-lbs. of energy during the explosion stroke and give it out during the compression stroke. Figure 314 shows a crank moment diagram of a four-cylinder 3½ × 5-in. engine for one-half revolution, this diagram repeating itself every half revolution. It will be seen that in this case the flywheel must store up 69 ft.-lbs. of energy during the first half of every stroke and give it out again during the last half. Figure 315 shows a crank moment diagram for a six-cylinder 3½ × 5-in. engine for two-thirds of a revolution. In this case the fly-

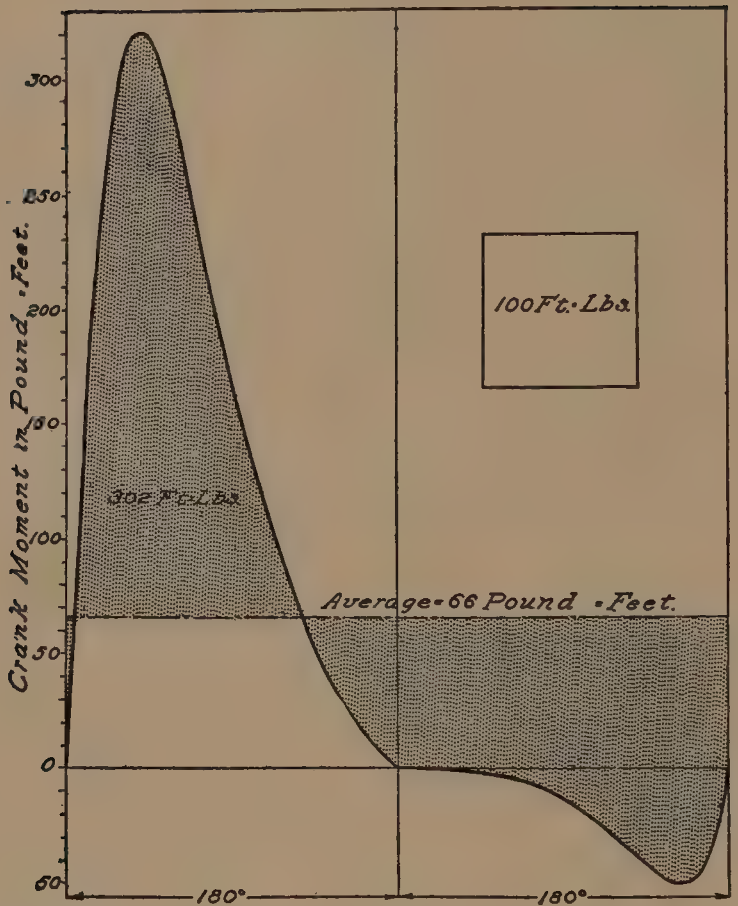
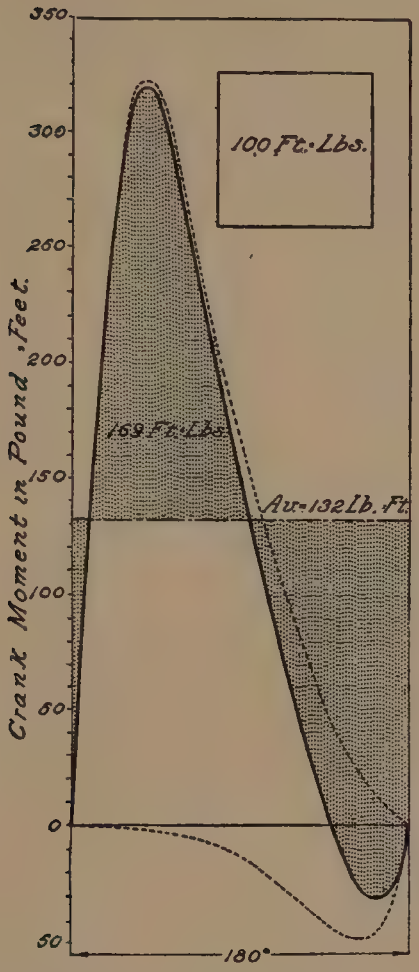


FIG. 313.—LOW-SPEED CRANK-MOMENT DIAGRAM FOR TWO-CYLINDER 3½ BY 5-IN. ENGINE.

wheel must store up and deliver periodically 82 ft.-lbs. The flywheel capacities required with engines of different numbers of equal sized cylinders for the same degree of regularity of operation therefore compare as follows:

Single cylinder	100 per cent
Double cylinder, equally timed...	80 per cent
Four cylinder	44 per cent
Six cylinder	22 per cent
Eight cylinder	11 per cent
Twelve cylinder	4 per cent

As a matter of fact, much greater flywheel capacities than those indicated are used on multi-cylinder engines, for the



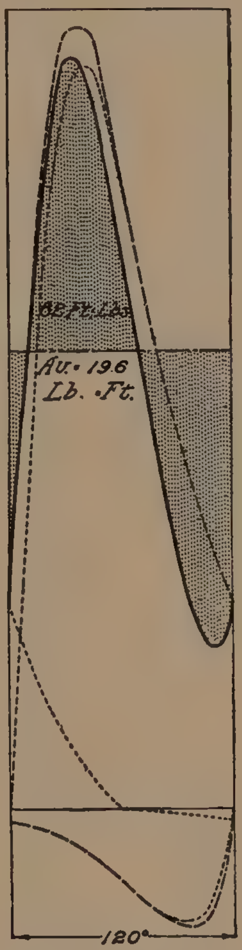


FIG. 314 (LEFT).—LOW-SPEED CRANK-MOMENT DIAGRAM FOR FOUR-CYLINDER 3½ BY 5-IN. ENGINE.

FIG. 315 (RIGHT).—LOW-SPEED CRANK-MOMENT DIAGRAM FOR SIX-CYLINDER 3½ BY 5-IN. ENGINE.

reason that a car with such an engine is expected to throttle down much lower than one with only a few cylinders.

In the case of large four- and six-cylinder engines that are started by hand there is another reason for making the flywheel somewhat larger than necessitated by running conditions. Without a flywheel it would be impossible for the operator of average strength to turn over the crankshaft of an engine of, say, 5-in. bore, against compression. The provision of a heavy flywheel enables him, when spinning the engine, to supply the requisite energy for compressing the charge at a practically uniform rate during two revolutions of the flywheel. The greater the flywheel capacity the less will be the fluctuation of the cranking resistance at a certain average cranking speed.

Energy Stored in Flywheel—The kinetic energy of a moving weight is

$$E = \frac{wv^2}{2g} \text{ ft.-lbs.}$$

when w is given in pounds; v, the speed in ft. per second and g the acceleration of gravity = 32.16 ft. per second per second. Now, referring to the sketch, Fig. 316, let w be the weight of a small particle of matter in the rim of a flywheel at a distance $r/12$ ft. from the center of rotation, and let the flywheel rotate with an angular speed of $n/60$ revolutions per second. Then the velocity of the particle w is evidently

$$2\pi\frac{r}{12}\frac{n}{60} \text{ feet per second,}$$

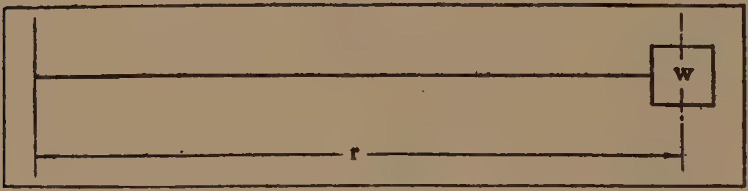


FIG. 316.

and the energy stored up by the particle w is

$$E = \frac{4w\pi^2r^2n^2}{2g \times 12^2 \times 60^2} = \frac{w\pi^2r^2n^2}{259{,}200g} \text{ foot-pounds.} \quad . \quad (93)$$

The angular speed of the engine

$$\omega = \frac{2\pi n}{60} \text{ radians per second,}$$

and substituting this in equation (93) we have

$$E = \omega^2\frac{wr^2}{288g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (94)$$

The expression $(wr^2)/(144g)$ or $(w/g)(r/12)^2$ is known as the polar moment of inertia (I_p) and is a measure of the flywheel capacity. In calculating this dimension, w is expressed in pounds; $r/12$ represents feet, and g is expressed in feet per second per second, hence the polar moment of inertia or flywheel capacity is expressed in feet-pound-seconds.[2] The stored energy may therefore also be expressed as follows:

$$E = \frac{I_p \omega^2}{2}.$$

In order to obtain an accurate expression for the energy stored in the entire flywheel, it is necessary to first develop an expression for the energy stored at a given speed in a cylindrical element of radius r and thickness dr and then integrate this expression between the limits r_0 and r_1. However, in practical work sufficient accuracy is obtained by considering the entire weight of the rim concentrated at a distance from the axis of rotation equal to the arithmetical mean between the inner and outer radii of the rim. The weight of the rim is

$$w = (r^2{}_0 - r^2{}_i)\,\pi a \times 0.26 \text{ pound},$$

and the mean radius is

$$\frac{r_0 + r_i}{2} \text{ inches.}$$

The linear velocity of a point at the middle of the rim is therefore

$$\pi \frac{n}{60}\left(\frac{r_0 + r_i}{12}\right) \text{ feet per second.}$$

After these two values have been found they are inserted in the formula for kinetic energy and the calculation is carried through. As a rule only the energy stored in the rim is calculated with, though, of course, the web also has some flywheel effect.

Dependence of Flywheel Capacity Required on Bore and Stroke—In the foregoing we investigated the flywheel capacities required in engines with different numbers of cylinders. We will now consider the problem of how the flywheel capacity should vary with the bore and stroke, the number and arrangement of the cylinders remaining the same. Take a single-cylinder engine, for instance. All the energy is developed during the power stroke, but the engine does work at a substantially uniform rate during all four strokes. The flywheel, therefore, must store up about three-quarters of the energy liberated by the explosion. In reality the fraction is a little higher, because during the compression stroke the flywheel must supply the energy not only for doing the outside work, but also for compressing the charge. The point is that a certain fixed percentage of the energy of each explosion is stored in the flywheel, and the fluctuation in its speed caused by the absorption and return of this energy must not exceed a certain percentage. Now, the energy of an explosion is directly proportional to the piston displacement or to the expression b^2l (b being the bore and l the length of stroke). We have already found an expression (94) for the energy storing capacity of the flywheel, and we may therefore write

$$b^2l \sim \omega^2 \frac{wr^2}{288g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (95)$$

But the angular velocity ω increases as the length of the stroke l decreases, and vice versa. It was formerly considered that the two factors varied in inverse ratio, but it is now generally admitted that a long stroke engine will run at higher piston speeds than a short stroke one, and it is therefore advisable to write

$$\omega \sim \frac{1}{\sqrt{l}}.$$

Inserting this value in proportionality (95) we have

$$b^2l \sim \frac{1}{l} \frac{wr^2}{288g},$$

$$b^2l^2 \sim \frac{wr^2}{288g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (96)$$

From this it follows that the flywheel rim weight

$$w = c\left(\frac{bl}{r}\right)^2 \text{ lbs.}$$

where c is a constant varying with the number of cylinders. An analysis of considerable data regarding flywheel dimensions on hand shows that the average value of c is 10.5 for single-cylinder high-speed engines, 9 for double-cylinder and 7.5 for engines with four or more cylinders.

The outside diameter of the flywheel is a function of the length of stroke and is generally made equal to three times the stroke. With unit power plants there naturally is an inclination to reduce the diameter, as this permits a saving on the weight of the flywheel housing. Another consideration is that the flywheel housing is often the lowest part of the car and increasing the flywheel diameter reduces the road clearance. Any moderate variation in the flywheel capacity does not appreciably affect the operation of the engine at normal speeds. An increase in flywheel capacity, however, enables an engine to operate better at low speed, thus increasing the flexibility and making it more difficult to stall the engine by letting in the clutch a little too rapidly.

A six-cylinder $3\frac{1}{2} \times 5$-in. engine would have a flywheel with a mean rim radius of $6\frac{1}{2}$ in., and the weight of rim required according to the rule developed in the foregoing would be

$$7.5\left(\frac{3.5 \times 5}{6.5}\right)^2 = 54 \text{ lbs.}$$

The clutch, which is generally built right into the flywheel, and the heavy crankshaft used on modern high-speed engines add materially to the effective flywheel capacity.

Centrifugal Force on Flywheel—When a flywheel revolves about its axis, every particle of matter in it is subjected to centrifugal force which acts radially outward. The value of this force has already been given as

$$F = 1.226wn^2r,$$

where n is the number of revolutions per second and r the radius of rotation in feet. It will, however, be better to continue to express n in revolutions per minute and r in inches, in which case we have

$$F_c = \frac{wn^2r}{35{,}240}$$

The weight w of the rim is approximately

$$2\pi rha \times 0.26 \text{ pound},$$

where h is the radial depth of the rim in inches. Inserting this value in the equation for the centrifugal force we have

$$F_c = \frac{\pi r^2n^2ha}{68{,}000} \text{ (appr.)}$$

This centrifugal force tends to rupture the rim in a plane passing through its axis. The component of the centrifugal force perpendicular to this plane is equal to one-half the total force multiplied by the ratio of the diameter to half the circumference

$$F_n = \frac{\pi r^2n^2ha}{68{,}000} \times \frac{1}{2} \times \frac{2r}{\pi r} = \frac{r^2n^2ha}{68{,}000}.$$

This tension is taken up by two cross-sections at opposite ends of a diameter whose combined area is $2ha$ square inches. The unit stress therefore is

$$S = \frac{\dfrac{r^2n^2ha}{68{,}000}}{2ha} = \frac{r^2n^2}{136{,}000} \text{ lbs. per sq. in.}$$

Taking the tensile strength of ordinary cast iron to be 16,000 pounds per square inch, a flywheel would burst when running at a speed

$$n = \sqrt{\frac{16{,}000 \times 136{,}000}{r^2}} \text{ r.p.m.}$$

The maximum value of r for automobile flywheels is about 9 inches, hence the bursting speed is

$$\sqrt{\frac{16{,}000 \times 136{,}000}{9 \times 9}} \doteq 5.180 \text{ r.p.m.}$$

Of course, if there are blowholes or similar defects in the casting it may burst at a lower speed.

Most engines seldom run at more than one-half this speed, under which condition the factor of safety would be 4, as the stresses increase with the square of the speed. The safety factor is further increased by the solid web generally used. A few manufacturers use flywheels of steel and of "semi-steel" (high tensile cast iron), probably as much because these make better materials for the starter gear ring as because of the additional safety afforded.

Fluctuations in Car Speed—In connection with the flywheel capacity of the engine the flywheel effect of the car itself when going at speed must be taken into account. It was assumed so far that the resisting moment on the engine was constant. This, however, is by no means the case when the car is traveling at a fairly high speed. It then has stored up a great deal of kinetic energy, upon which it draws immediately the engine slows down.

Calculation shows that with an engine of only four cylinders, in high gear and at normal speed, the fluctuations in car speed do not attain one-tenth of 1 per cent, whereas when pulling hard up hill in low gear the fluctuations may be 10 per cent or more.

Rim and Web—The flywheel generally serves as one member of the friction clutch, and in most passenger cars it also serves as the driven gear of the starter drive, and these two auxiliary functions determine its shape to quite an extent.

From the standpoint of minimum weight for a given flywheel capacity, as much of the mass as possible should be located near the outer circumference, but while the flywheel "efficiency," as it might be called, is thus increased, the factor of safety is reduced. With steel it is easy to secure an adequate factor of safety, and the flywheel may then be designed for minimum weight, as indicated in Fig. 317 (Wills eight-cylinder engine). A cast-iron flywheel designed more with a view to safety than to high flywheel efficiency is illustrated in Fig. 318. In both cases the teeth for the starter drive are cut directly in the flywheel rim. In the case of the steel flywheel a housing for the clutch is bolted to the rim, while in the case of the cast-iron flywheel the rim itself forms this housing, one of the clutch members being bolted to the rear side of same. Single disc clutches are being used more and more, which are naturally of a somewhat larger diameter than multiple disc clutches, and must be of an outside diam-

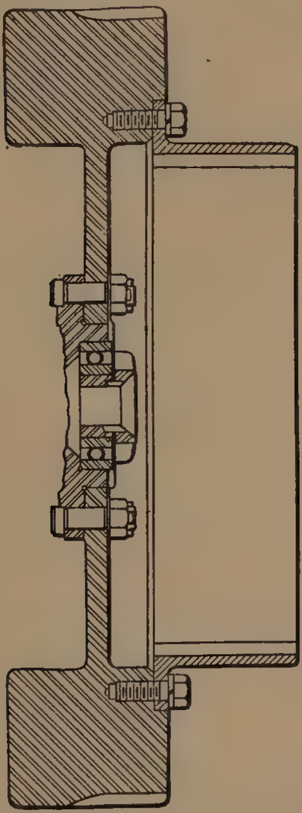


Fig. 317.—Steel Flywheel.

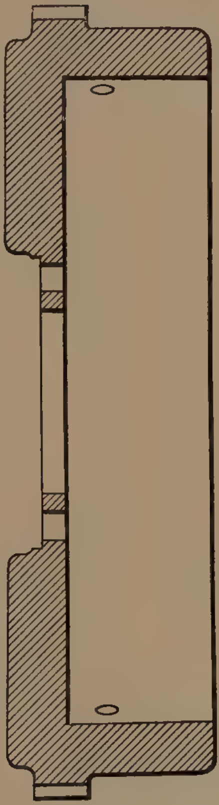


Fig. 318.—Cast-Iron Flywheel.

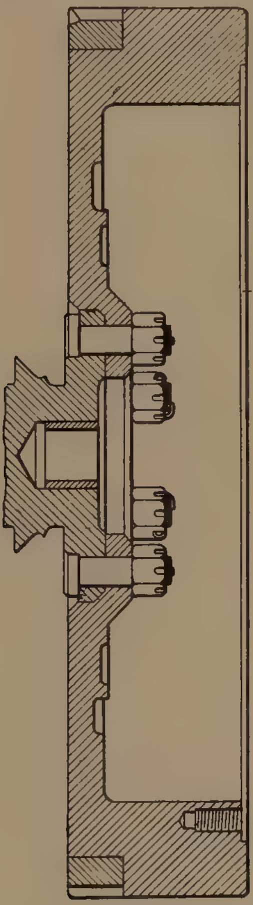
eter about the same as the inside diameter of the flywheel rim would naturally be made. When a multiple disc clutch is used, and especially with a fairly low-speed four-cylinder engine, it is common to use a separate housing, smaller than the inside diameter of the flywheel rim, and bolted to the web. With a cone clutch the flywheel rim usually forms the driving member of the clutch, and its rim is turned out to an angle of about 15° with the flywheel axis.

Fastening to Crankshaft—Some of the higher grade engines have starter gear rings of steel, which are shrunk on the cast-iron flywheel. This is illustrated in Fig. 319, which shows the flywheel of the Packard eight. Flywheels are generally secured to flanges integral with the crankshaft by means of six studs or bolts with properly locked nuts. However, when it is desired to use a ball bearing at the rear end of the crankshaft, or if the timing gear is located at the rear and a gear pinion or sprocket must be secured to the crankshaft at this end, this is impossible. It is then customary to use a steel center which is keyed to a tapered seat on the crankshaft and to which the flywheel is bolted the same as to an integral flange. This is illustrated in Fig. 320.

When gasoline engines were still built mainly in single and double cylinder types there was much opposition to the practice of keying flywheels, because the crankshafts were generally of comparatively small diameter and the sudden impulses at the moments of explosion when running at low speeds tended to loosen the keys, but with the large diameter shafts now used, the very considerable inertia of the crankshaft itself, and the comparatively uniform torque of the multi-cylinder engine the objections to this practice are no longer serious. Of course, a separate steel center complicates the design, and it is used only when there is need for it.

Machining of Flywheels—Flywheels are generally machined in multi-station chucking and turning machines, such as the Bullard Contin-U-Matic. This latter machine is adapted to three different types of tooling, known respectively, as the single-unit, two-unit and four-unit types. With single-unit type tooling identical operations are performed at all of the four stations of the machine, and four pieces are produced during each complete cycle of the carrier. With two-unit tooling, operations at alternate stations are the same; for instance, roughing may be done at stations 1 and 3 and finishing at stations 2 and 4. The piece which has been roughed at station 1 is taken off and re-chucked at station 2, thus going twice through the cycle of the carrier, and in operation the machine completes two pieces per cycle.

The single-unit type must usually be used in batteries of at least four, in order that the four chuckings required on a flywheel (roughing and finishing each side) may be completed by this method. The two-unit type may be installed in batteries of two, one for roughing and finishing each side; or, again, a roughing machine may rough both sides, and a finishing machine finish both sides. However, the machining times for roughing and finishing are not the same, and for this reason in many cases three or four roughing machines are balanced in production by one or two finishing machines.



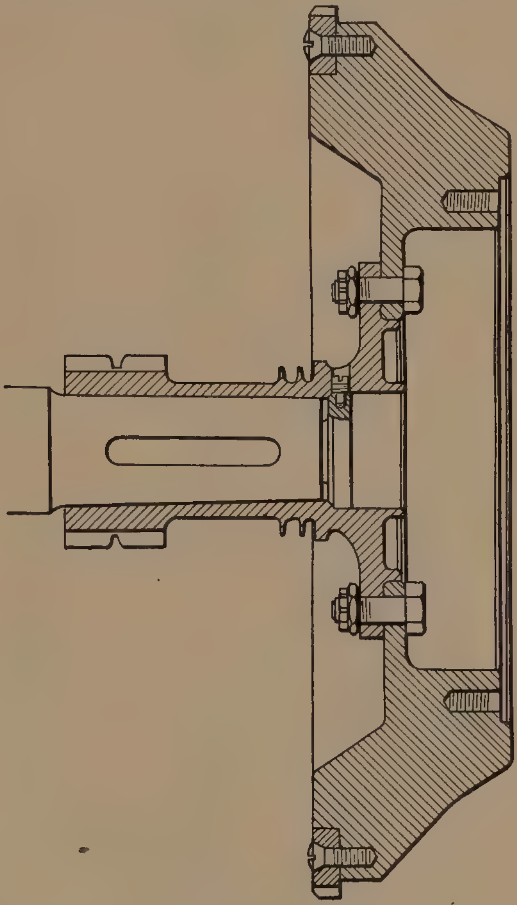


Fig. 319 (left).—Flywheel with Shrunk-on Steel Starter Gear Ring.

Fig. 320 (right).—Flywheel with Keyed Steel Center.

Successive operations on a flywheel in a Contin-U-Matic with four-unit type tooling is illustrated in Fig. 321. This wheel is roughed at Station 1, roughed on the reverse side

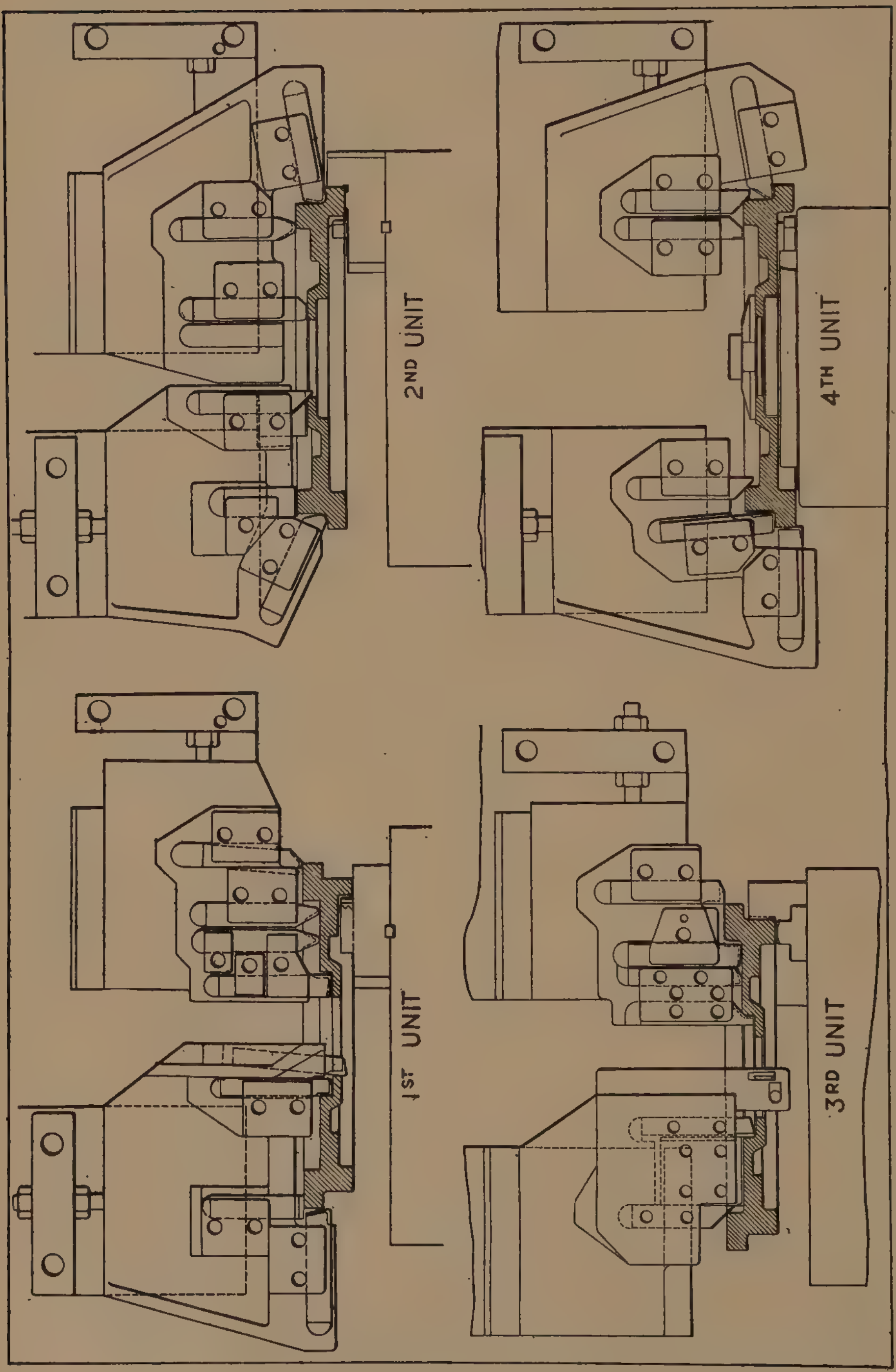


Fig. 321.—Sequence of Machining Operations on a Flywheel in a Bullard Contin-U-Matic.

at Station 2, finished at Station 3 on one side and finished on the other side at Station 4. Thus the flywheel is completely machined in this one machine. The flywheel is 15 in. in diameter; it is revolved at 15.5 r.p.m. and the cutting speed at the rim is about 60 ft. p.m. A production of 139 flywheels in 8 hours, or 15.5 per hour, is claimed, at an operating efficiency of 85 per cent. This method does not take advantage of the shorter time required by the finishing operation, but it has the advantage that the units are independent and the number of machines in operation can be cut down or increased as the production schedule requires.

In Fig. 322 is shown a Bullard Contin-U-Matic with four-unit type tooling in which operations on the flywheel at Station 2 have just been completed.

Balancing Flywheels—The flywheel is that rotating part of the engine which has the greatest polar moment of inertia, and unless it is carefully balanced it may cause serious vibration of the engine and car frame at high speeds. It is not sufficient that it be in static balance; that is, that it will remain in any position when mounted on a shaft and put on balance ways; it must also be in running or dynamic balance. Of course, the great majority of flywheels are machined all over and therefore should be in complete balance, theoretically, but the density of cast iron is never quite uniform, and in large castings there will always be porous spots and blowholes, which throw them out of balance even if machined very accurately.

A special machine for balancing flywheels is manufactured by the Tinius Olsen Testing Machine Co., and is illustrated in Fig. 323. It is in the form of a box-like base or housing within which is contained most of the mechanism. The end of a vertical shaft will be seen projecting from the housing on top. This is fitted with an "adapter" for flywheels of any particular size of center bore.

The structure which carries this shaft has two degrees of freedom; that is, it is capable of vibrating around two axes at right angles to each other. One of these axes is determined by a pair of knife edges and passes through the axis of the vertical shaft, near the top of that shaft, parallel with the axes of the three hand wheels on the front of the machine. The supporting structure being locked against vibration around its other axis by means of the handle seen at the right near the top of the machine, the flywheel is spun around its axis by an electric motor, and if it is out of balance it will set the machine vibrating around the knife edge axis. Then, by means of the central handwheel, the shaft carrying the flywheel is moved up or down until the vibration ceases, which it does when the plane of unbalance is at the same level as the axis of the knife edge support. Any vibration is indicated by standard dial gauges, of which there are two on the front of the machine, near the right-hand top corner, one for vibration about each axis.

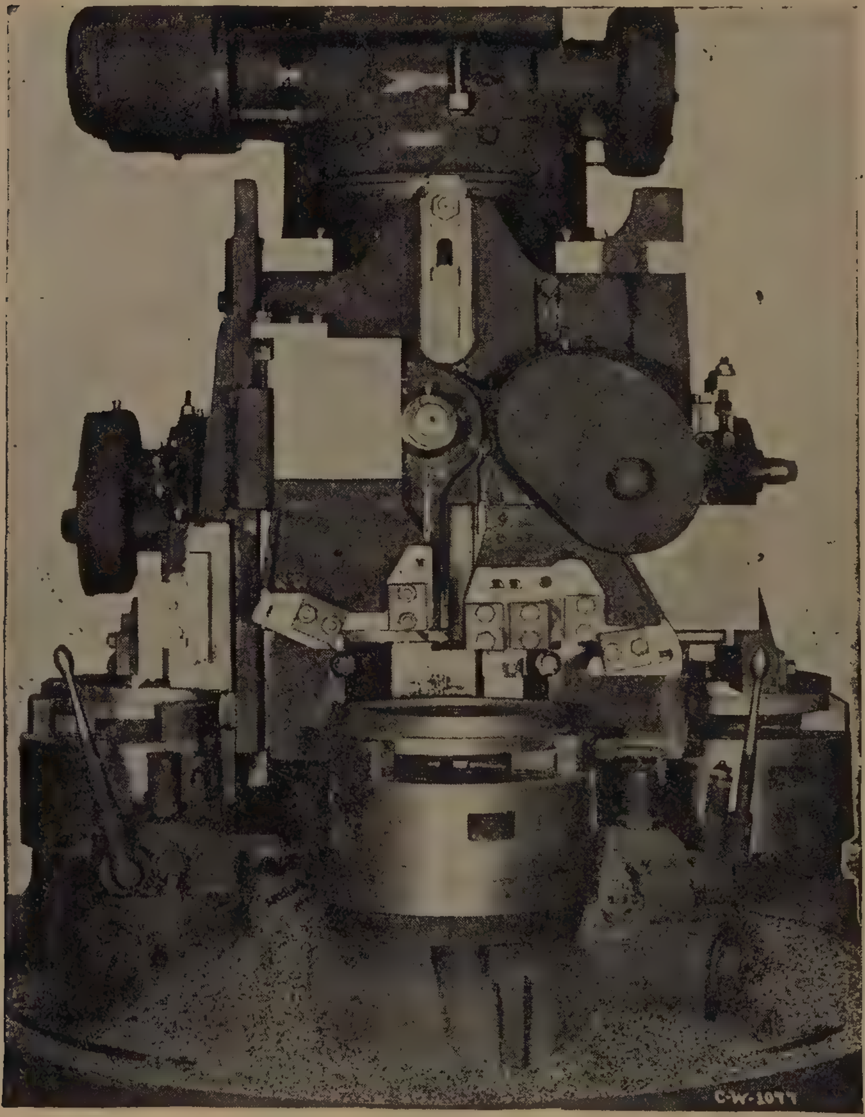


Fig. 322.—Bullard Contin-U-Matic, Showing Flywheel at End of Operation in Station 2.

When vibration ceases, the axial plane of unbalance has been determined, being indicated by the pointer at the right. Next the supporting structure is locked against vibration around the knife edge axis and freed so as to be capable of vibration around a vertical axis, by means of the handled latch near the dial gauges. The flywheel is now spun again and will vibrate around the vertical axis. From the indication of the gauge the operator can fairly accurately judge the amount of unbalance, and by means of the handwheel on the right he then introduces an artificial moment, equal to what he considers the moment of unbalance to be, the effect produced being the moving of two rotatable weights, located on opposite sides of their common axis of rotation, axially and in opposite directions.

The vibration will be either increased or decreased, according to whether the angle between the unbalanced moment of the flywheel and that created artificially is less or more than 120°. From the effect on the amplitude of vibration as indicated by the dial gauge, it is possible to judge the angular relation between the two moments, and by means of the handwheel at the left the artificial moment can then be moved angularly until it is in direct opposition to the unbalanced moment of the flywheel. It is not really necessary to judge the angle between the two moments at the beginning, because as soon as the left-handwheel is moved, the vibrations indicated by the dial will decrease or increase, thus showing whether the point of balance is being approached or moved away from. If the operator's judgment was correct as to the amount of unbalance, vibration will cease; in any case, it will be greatly reduced. A finer adjustment for both amount and angle of unbalance can then be made by means of the two handwheels. If either of the handwheels is moved in the wrong direction the gauge will at once indicate increased vibration, and vice versa, hence the adjustment resulting in the complete elimination of vibration is quickly found, and the moment of unbalance and its angular position can then be read off on dials back of the two handwheels. The limit of accuracy of the machine is said to be ⅛ oz.-in.

With this machine from 12 to 15 flywheels can be balanced per hour, according to the manufacturer. The axial and radial planes of unbalance and its amount being determined, the flywheel is corrected by drilling holes at the proper point, removing the necessary amount of metal.

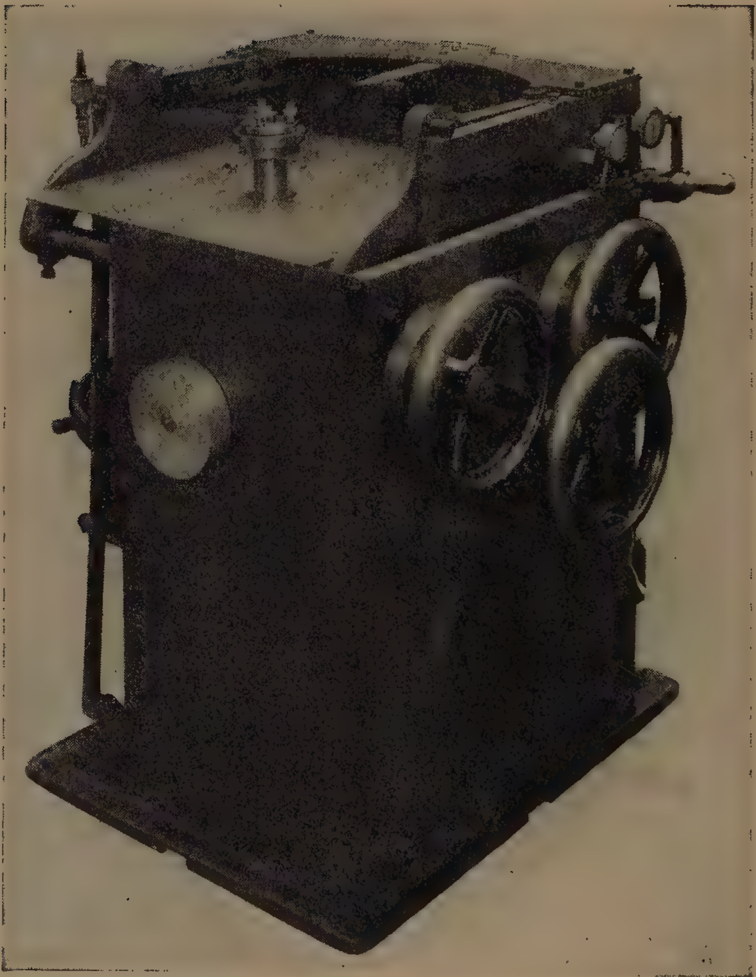


FIG. 323.—OLSEN FLYWHEEL BALANCING MACHINE.

CHAPTER XVIII

Speed Control—The Governor

The speed of an automobile engine is controlled by means of a throttle valve in the carburetor, which is generally of the butterfly type, but occasionally—especially on foreign cars —of the barrel type. This valve is connected to a finger lever located on the steering wheel or underneath it on the steering column, and frequently also to an accelerator pedal, so that the driver can vary the rate of mixture supply to the engine at will, and hence control its speed.

Generally the linkage is so arranged that the throttle valve can be opened to any degree by the accelerator pedal without disturbing the position of the throttle lever on the steering wheel. The accelerator pedal is retracted by a spring, and if the driver lifts his foot off the pedal, the throttle automatically closes or returns to a position determined by the setting of the lever. The latter has either a friction or a ratchet lock, and thus holds the throttle in position until its setting is changed.

Whenever the load is decreased the engine speeds up, and this tendency to ''race'' is especially strong when the clutch is disengaged and, therefore, all load removed. In order to prevent the engine from attaining excessive speeds under these conditions, the throttle must always be closed immediately. Careless drivers often disregard this rule, thereby injuring their engines, and for this reason the engines of commercial vehicles, especially those of larger size, are generally fitted with governors which automatically maintain the speed of the engine within a certain narrow range. The engines of farm and industrial tractors also are fitted with governors, as a rule.

There are three general types of governor in use on automotive engines, viz., the gas inertia type, the centrifugal type and the hydraulic type. The centrifugal type is the one most widely used and appears in a variety of designs. We will discuss these different types in succession. Whichever type is used, it is connected to the throttle valve in such a manner that as soon as the speed for which it is set is exceeded, it closes the throttle valve more or less, so as to keep the speed within the predetermined limit. In addition to the automatic control of the engine, some form of hand or foot control is sometimes provided. The governor works against a spring, and the hand or foot control mechanism is so arranged that it varies the position of the spring rest, in which case the governor will hold the engine at any speed the operator may desire.

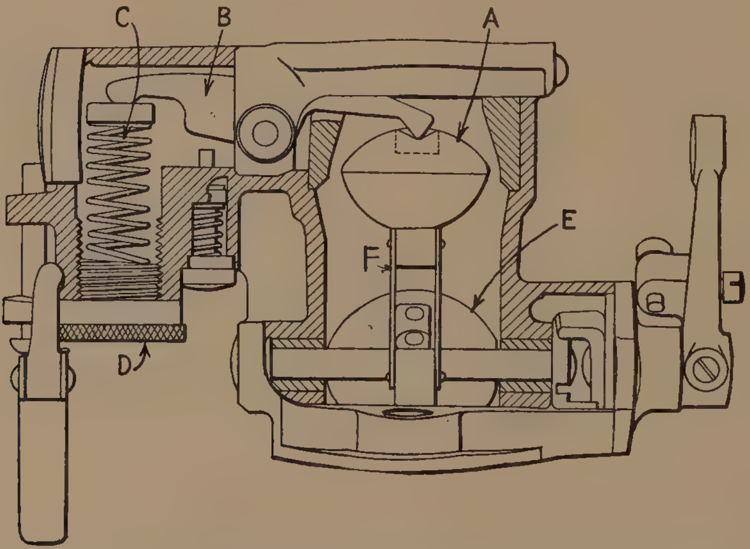


Fig. 324.—Gas-Inertia-Type Governor (Monarch).

Gas Inertia Type—A current of air or gas has a tendency to carry along with it any object in its path, with a force proportional to the square of its speed. This principle is made use of in the gas-inertia type of governor. An example of this type is the Monarch, illustrated in Fig. 324. A body A known as a conoid is located inside a conical section of the inlet tract. In the upper part of the housing there is a double-armed lever B, one end of which is pressed against by the governor spring C while the other end contacts with the conoid A. Adjustment is effected by removing the cap D and turning a slotted screw provided for this purpose. The lever B is provided at both ends with hardened ball-point bearings that oscillate in hardened steel cups, and it has an oil-wick lubricated bronze bushing.

Normally the conoid is held in its lowest position by the governor spring, and the butterfly throttle E, to which it is connected by a pair of links F, is then wide open. When the gas speed past the conoid exceeds a certain value, the pressure of the gas against the conoid overcomes the pressure of the spring on the latter, the conoid is drawn up into the cone and the throttle is partly closed. As with all pneumatic or gas-inertia-type governors, throttle and governor form a single unit.

Centrifugal Governors—The centrifugal governor, in the form in which it is most frequently used, consists of two weights carried upon one of the revolving parts of the engine, generally the camshaft. The weights are carried by bell cranks which are pivoted on brackets rotating with the shaft and generally either formed integral with or fixed to the camshaft gear. The free arms of the bell cranks engage with a grooved sliding sleeve on the camshaft. This sliding sleeve is provided with a second groove into which engages the end of a lever or bell crank whose other end connects to the arm of the throttle valve. The revolving weights are being drawn or forced together by means of a spring or springs which may either extend between them or surround the shaft and exert their pressure against a stop collar and the sliding sleeve respectively. Fig. 325 shows a centrifugal governor in diagram. We found that the centrifugal force on rotating bodies is

$$F_c = \frac{wn^2r}{35,240}.$$

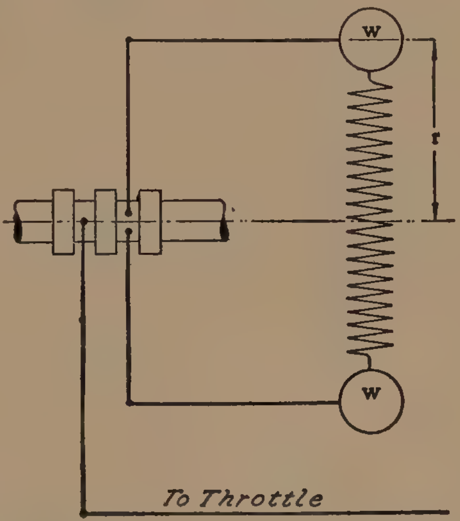


Fig. 325.—Diagram of Centrifugal Governor.

In the case of the governor this force is balanced by the spring force in any position of equilibrium. The law of the coiled extension spring is that its pull is proportional to the extension. Let a be the length of the spring with its connections in the free state. Then $2r - a$ is the extension, and the spring force is

$$F_s = c(2r - a).$$

Hence

$$\frac{wn^2r}{35{,}240} = c(2r - a)$$

$$35{,}240ca = (70{,}480c - wn^2)r$$

$$r = \frac{35{,}240ca}{70{,}480c - wn^2}. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (97)$$

In the above w is taken as the weight of one of the governor weights, since the spring tension is equal to the centrifugal force on one of the weights only.

It will at once be seen that r increases with the speed n. The speed at which the governor acts is, therefore, necessarily higher for small throttle openings or light engine loads than for large throttle openings or heavy loads, but the difference between the two speeds need not be very great.

Now let r_1 and r_2 be the smallest and greatest radii of rotation, respectively, of the governor weights and n_1 and n_2 the corresponding speeds of balance. Then, from equation (97),

$$70{,}480r_1c - wn_1^2r_1 = 70{,}480r_2c - wn_2^2r_2$$

$$70{,}480(r_2 - r_1)c = w(n_2^2r_2 - n_1^2r_1)$$

$$c = \frac{w(n_2^2r_2 - n_1^2r_1)}{70{,}480(r_2 - r_1)} \text{ pounds per inch.} \quad . \quad . \qquad (98)$$

also

$$r_1 = \frac{35{,}240ca}{70{,}480c - wn_1^2}$$

$$70{,}480cr_1 - wn_1^2r_1 = 35{,}240ca,$$

$$a = \frac{70{,}480cr_1 - wn_1^2r_1}{35{,}240c}$$

$$= 2r_1 - \frac{wn_1^2r_1}{35{,}240c}$$

$$= 2r_1 - \frac{wn_1^2r_1}{\dfrac{35{,}240w(n_2^2r_2 - n_1^2r_1)}{70{,}480(r_2 - r_1)}}$$

$$= 2r_1 - \frac{2n_1^2r_1(r_2 - r_1)}{n_2^2r_2 - n_1^2r_1}. \quad . \quad . \quad . \qquad (99)$$

In the above, c represents the tension or force required to extend the spring 1 inch in length and a the length of the spring in the free state, inclusive of the hooks or connections to the centers of the weights. Now let it be desired that the governor act at 1000 r.p.m. of the engine or 500 r.p.m. of the camshaft when the throttle is fully open, and at 1200 r.p.m. of the engine or 600 r.p.m. of the camshaft when the throttle is near the minimum opening position. The limiting radii of rotation of the governor weights may be assumed to be 1.5 and 2 in. respectively, and the weight w of each governor ball 1 lb. Substituting in equation (98)

$$c = \frac{1(600^2 \times 2 - 500^2 \times 1.5)}{70{,}480(2 - 1.5)} = 9.8 \text{ lbs. per inch,}$$

and in equation (99)

$$a = 2 \times 1.5 - \frac{2 \times 500^2 \times 1.5(2 - 1.5)}{600^2 \times 2 - 500^2 \times 1.5} = 1.92 \text{ in.}$$

The combined length of the connections from the ends of the spring proper to the center of the weights can be kept within 0.42 in., in which case the length of the spring in its free state should be 1.5 in. If there are to be two springs, one at either side of the weights, then the force required to extend either of them a distance of 1 in. should be

$$\frac{9.8}{2} = 4.9 \text{ lbs.}$$

The problem then consists in designing a helical spring which in its free state measures 1.5 in. in length, which will extend at the rate of 1 in. per 4.9 lbs. applied to its ends and which will not be overstrained by extending it to

$$4 - 0.42 = 3.58 \text{ in.}$$

This can be done by means of the rules for the design of coiled springs which were given in the chapter on Valves and Valve Gears. If the springs turn out to be of unsuitable size, the assumptions regarding the range of motion of the governor weights or the speed range through which the governor acts can be varied and the calculation made over again.

Hunting—Suppose the governor weights to occupy a certain position intermediate between the two limiting positions, and to be held in equilibrium by the centifugal force acting outwardly and the tension of the springs acting inwardly. Now, let the speed of the engine be gradually increased. This, of course, will increase the centrifugal force, and there is then a tendency for the weights to fly out from the axis of rotation. However, there will be no actual outward motion until the excess of centrifugal force over the spring force is sufficient to overcome the static friction of the throttle and of the linkage between it and the governor weights. Hence the engine will gain some speed before the governor begins to act. The moment the speed becomes great enough for the centrifugal force to overcome both the spring tension and the frictional resistance, the governor weights fly out and the throttle begins to close. Now, as soon as the throttle and its linkage begin to move, their resistance to motion is somewhat reduced, according to the well-known law that the friction of rest is

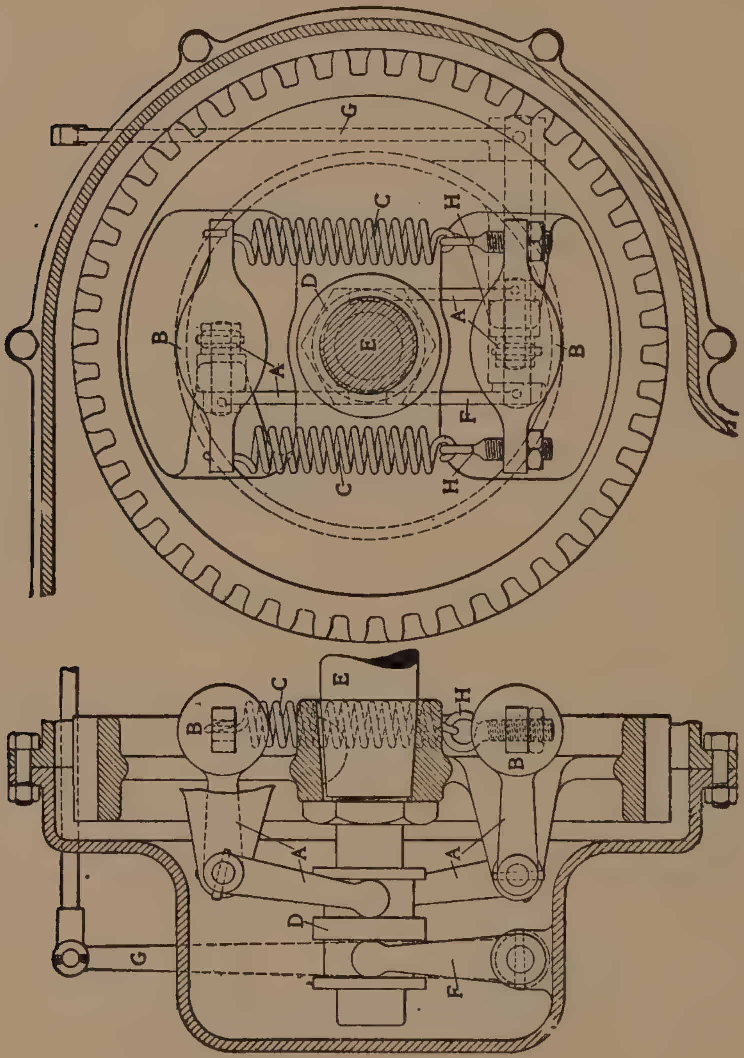


FIG. 326.—CENTRIFUGAL GOVERNOR.

greater than the friction of motion. The result is that the throttle will be closed further than it really should in order to keep the speed of the engine constant. The engine will thereby be throttled down in speed, and the throttle in turn will be opened wide by the governor. In this way the throttle may be opened and closed by the governor and the engine in consequence speed up and slow down periodically, a phenomenon known as "hunting." Hunting is particularly likely to occur when starting the engine, when its joints are stiff. It can be reduced to a minimum by careful fitting so that the friction at the bearings of the throttle shaft and at the joints in the linkage is very small, and by carefully balancing the throttle valve.

Types of Centrifugal Governor—The typical form of shaft governor as used on automobile engines is illustrated in Fig. 326. Two bell cranks AA carrying the governor weights BB are pivoted to brackets extending from the web of the cam gear. The governor weights are connected by helical springs CC secured to them by hooks at opposite ends. The inwardly extending arms of the bell cranks engage into a groove on the sliding sleeve D on the camshaft E. Into a second groove on this sliding sleeve engages the end of a lever arm F whose shaft extends through and has a bearing in the wall of the cam gear housing. This shaft at its end outside the housing carries a long lever G from which connection is made to the throttle arm. The lengths of lever G and the throttle arm are so proportioned that the range of throttle motion corresponds to the range of the governor. In the figure provisions for adjusting the tension of the governor springs CC are shown, in the shape of hook screws HH, which screw through the projecting ends of the governor weights BB and are locked in place by means of check nuts.

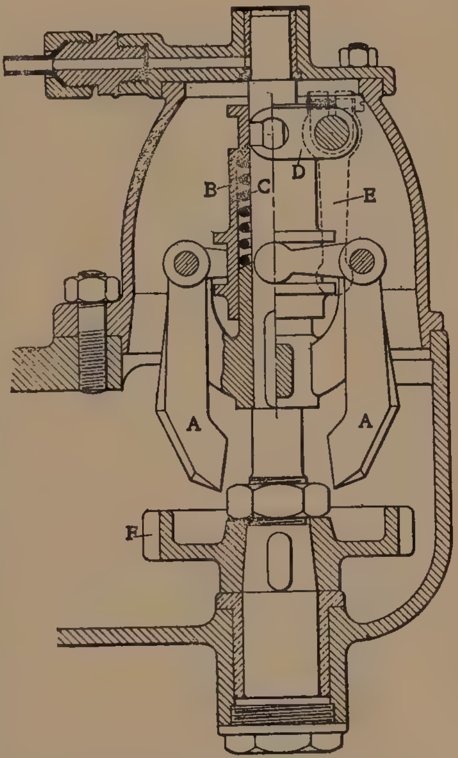


Fig. 327.—Vertical Type of Centrifugal Governor.

This governor is enclosed with the cam gears and requires no special housing.

Figure 327 shows a vertical type of shaft governor. The shaft may be extended at the top and bottom to drive such accessories as the timer and the oil pump, if desired. The governor comprises two pivoted weights AA developed in the form of a bell crank, whose operating arms engage with a grooved sleeve B. The counter spring C is located between this sleeve and the shaft. The sliding sleeve B is also in working connection with the lever D. The shaft of this lever extends through the wall of the housing and outside the housing carries a lever arm E from which connection is made to the throttle valve. The governor shaft receives its motion from the camshaft through a pair of helical gears, the driven one of which is shown at F. Lubrication of the governor is provided for by a direct oil feed to the top bearing.

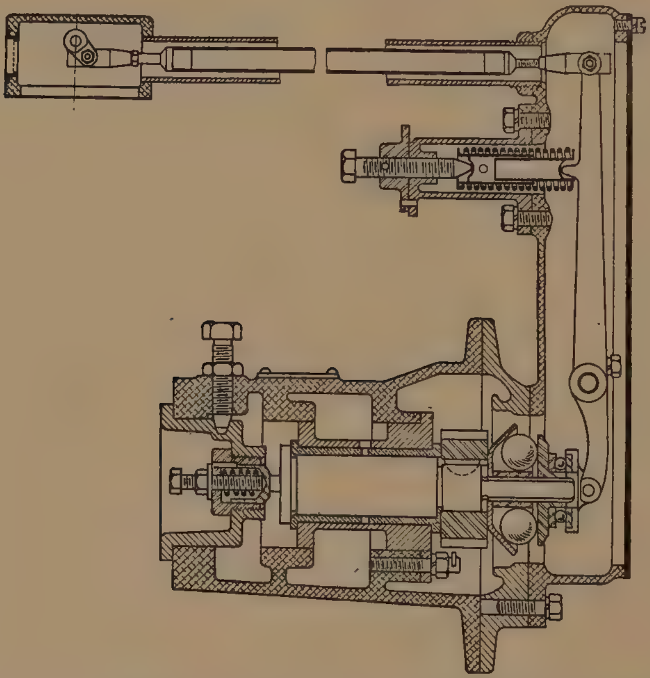


Fig. 328.—Steel-Ball Type of Centrifugal Governor.

In recent years a modified form of centrifugal governor has made its appearance, making use of steel balls held between a flat and a conical disc on the governor shaft. This is illustrated in Fig. 328. The adjusting means, in the form of a set screw, is most accessible, and the adjustment can be readily sealed against tampering.

Hydraulic Governors—A number of manufacturers, including Panhard & Levassor in France and the Packard Motor Car Company in this country, have used so-called hydraulic governors for governing the speed of their engines. In these the pressure of the pump circulated cooling water is made use of to close the throttle when the speed exceeds certain limits. A sketch of such a governor is shown in Fig. 329. The device consists merely of a diaphragm chamber A in which there is a diaphragm B acting against the plunger C from which connection is made to the throttle arm. The cooling water as it comes from the pump may be led through the diaphragm chamber, or this chamber may simply be connected by a pipe to the cooling system at a point near the pump outlet. The pressure of the circulating water increases with the speed of the pump, and hence, with the speed of the engine, the relation between the pressure and pump speed depending upon the type of pump employed. The throttle can be held open by means of a spring.

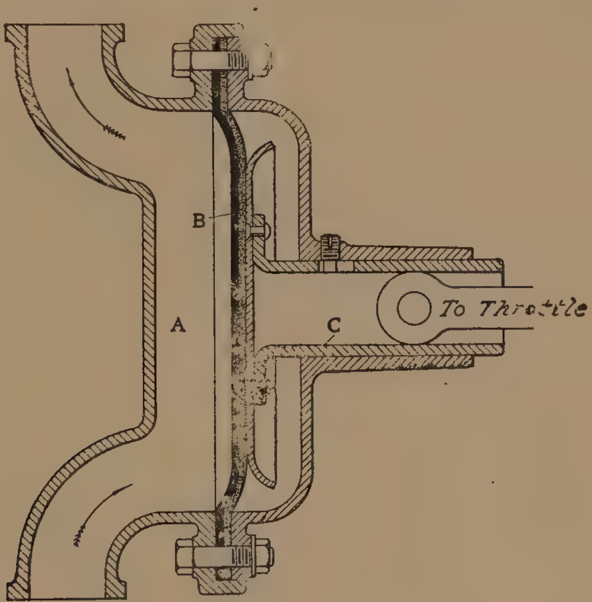


Fig. 329.—Hydraulic Governor.

CHAPTER XIX

Power Output and Other Characteristics

Many efforts have been made to develop a simple formula giving the maximum power output of gasoline engines, but these attempts have been more or less futile, owing to the very large number of factors which affect this output. The elements of design which determine the maximum output may be enumerated as follows:

Number of cylinders.
Bore.
Stroke.
Compression ratio.
Form of compression chamber.
Valve timing.
Valve port opening.
Cross-section, length and relative straightness of gas passages.
Cooling facilities.
Weight of reciprocating parts.
Location of ignition points.

Besides the above, the fit and alignment of the bearings, their relative diameters and lengths, and the amount and grade of lubricant used upon them affect the output at the crankshaft to some extent. It would evidently be impossible to embody all of these factors in a power output formula, and for this reason all except the number of cylinders and the bore and stroke are generally neglected.

Horse Power Formulae—If we designate by p the mean gas pressure per square inch of piston head area during the power stroke, reduced by the mean pressure during the compression stroke, and if the cylinder has a bore b and the length of stroke is l, then the work done in one cylinder during one cycle is evidently

$$\frac{\pi b^2 pl}{4 \times 12} \text{ ft.-lbs.}$$

If the engine runs at n revolutions per minute, then there will be $n/2$ explosions in each cylinder per minute, and the power developed by each cylinder will be

$$\frac{\pi n b^2 p l}{2 \times 4 \times 12 \times 33{,}000} \text{ H.P.}$$

Also, if there are N cylinders, then the power of the engine will be

$$\text{H.P.} = \frac{\pi N n b^2 p l}{3{,}168{,}000}.$$

This is the indicated power—the power conveyed by the expanding gases to the moving pistons. Of this power a certain percentage is lost in piston and bearing friction, valve operation, windage, etc. Representing the mechanical efficiency of the engine by η, we have for the brake horse power

$$\text{B.H.P.} = \frac{N n \pi b^2 l \eta p}{3{,}168{,}000}. \quad . \quad . \quad . \quad . \quad . \qquad (100)$$

The piston displacement is

$$D = \frac{N \pi b^2 l}{4},$$

and if we substitute this in the above we get

$$\text{B.H.P.} = \frac{n D \eta p}{892{,}000}.$$

Of these factors only D is known in advance; n, the speed at which the engine develops its maximum power, and $\eta\, p$, the brake mean effective pressure, are not known. If we desire a formula for the horse power involving only the bore and stroke, then the question arises whether the piston speed of maximum output and the mean effective pressure vary with the bore and stroke, and if so, in what way.

European Data and Deductions—An investigation of these questions was made by a committee of the Institution of Automobile Engineers (Great Britain). As regards the mean effective pressure, it would be expected that this would increase with the bore, for the reason that in a large-bore engine there is less cooling surface to the combustion chamber per unit of charge contained, hence less heat will be lost through the cylinder walls. This view is undoubtedly correct in so far as engines of the same compression ratio are concerned, and some experimental data were cited by the committee to support it. On the other hand, the very fact that there is a relatively larger wall surface in a small bore cylinder, and hence more rapid cooling of the charge, permits of using a higher compression ratio. A higher compression ratio would make up for the lesser area of cooling surface, so that the mean effective pressure would be substantially the same in all sizes of cylinder. This is borne out by a diagram in which the mean effective pressures of 88 engines were plotted against the cylinder bores, which accompanied the report of the committee. This diagram showed absolutely no variation of the mean effective pressure with the bore. Of 124 engines regarding which data were collected by the committee, 5.6 per cent showed a mean effective pressure above 90 lbs. per sq. in.; 15.3 per cent between 80 and 90 lbs., 28.2 per cent between 70 and 80 lbs., and 21.2 per cent between 65 and 70 lbs. About 50 per cent of the engines thus showed mean pressures over 70 lbs. per sq. in. The highest pressure found was 95½ lbs.

While these determinations were made a good many years ago, they substantially reflect conditions with respect to mean effective pressures obtaining to-day, for improvements resulting from refinements in design have been negatived by deterioration in the character of the fuel marketed, with respect to its ability to withstand high compression.

N. A. C. C. Rating Formula—In this country and England a rating formula which is known here as the N. A. C. C. and there as the R. A. C. formula, has long been used. It is as follows:

$$\text{B.H.P.} = \frac{Nb^2}{2.5}, \quad \ldots \ldots \quad (101)$$

at 1000 ft. piston speed per minute.

When this formula was first adopted, about 1905, most up-to-date engines gave their maximum output at about this piston speed, which is now greatly exceeded by all passenger-car engines of modern design, and even more by racing engines. In the case of heavy truck and other heavy duty engines the piston speed has not been increased so much. From equations (100) and (101) we have

$$\frac{Nb^2}{2.5} = \frac{Nn\pi b^2 l\eta p}{3{,}168{,}000}.$$

Since the piston speed

$$2nl/12 = 1000,$$

substituting this value in the right-hand member of the above equation we get

$$\frac{Nb^2}{2.5} = \frac{1000 \times \pi N b^2 \eta p}{528{,}000},$$

from which it follows that

$$\eta p = 67.2 \text{ lbs. per sq. in.}$$

which is the assumption on which the N. A. C. C. formula is based.

In 1912 the Technical Committee of the Automobile Club of France in a report to the French Government, recommended horse power formulae based on the following brake mean effective pressures: Passenger-car engines running at not over 1000 ft. piston speed per minute, 75.2 lbs. per sq. in.; high-speed passenger-car engines, 78.1 lbs.; truck engines, 82.4 lbs.

Dependence of Output on Bore and Stroke—In the opinion of many engineers, within reasonable limits the maximum output obtainable from a cubic inch displacement decreases as the size of the cylinder increases. There is evidently a lower limit to the range of cylinder sizes for which this holds true, but this is smaller than the cylinders usually employed in passenger-car practice. Support is lent to this view by the fact that in recent races in which the piston displacement has been limited to 122 cu. in., practically all of the designers used as many as eight cylinders, evidently believing that with this they could obtain a greater output than with any smaller number. This would indicate that the above-mentioned rule holds true at least down to cylinders of 15 cu. in. displacement.

If the stroke of the engine remains constant and the bore is increased, one would naturally suppose that the engine output would increase with the piston head area or with the square of the bore. The valves can be increased in size in direct proportion to the bore, hence at the same speed of revolution the gas speed through the valve ports would be the same in the larger as in the smaller engine, and there is therefore no reason to suppose that the larger bore engine cannot run at substantially the same speed. The reciprocating and rotating weights, of course, will be greater, but there also will be larger bearing surfaces to carry the loads due to the inertia and centrifugal forces on these weights.

Since the output increases as the square of the bore, but not as rapidly as the displacement of the cylinder, it follows that it increases less rapidly than the stroke. At one time the view was widely held that the output of an engine was practically independent of the stroke and depended only upon the bore; that a 4 by 6-in. cylinder, for instance, would give no more power than a 4 by 4-in., as it was supposed that the speed of maximum output would be 50 per cent greater with the latter than with the former, the piston speeds then being the same for both.

However, it has been shown that other things being equal, with a longer stroke a greater piston speed is attainable. This results from the fact that the inertia force due the reciprocating parts increases directly as the length of the stroke but as the square of the speed of revolution. Therefore, if one engine had a stroke 1.5 times as long as the other and ran at a speed two-thirds as great, the piston speeds of both would be the same, but the inertia forces in the long stroke would be only two-thirds as great as those in the short stroke engine. The inertia forces would be the same only if the speed of the long stroke engine were $\sqrt{2/3} = 0.816$ that of the other. From this point of view it would seem that the maximum output should vary as the square root of the stroke. The problem is complicated, however, by the fact that besides inertia forces there are a number of other factors which may possibly limit the output. Horse power output formulae have been proposed according to which the power depends upon the 0.6 and the 0.7 power of the stroke.

Dependence on Compression Ratio—The effect of the compression ratio on the power output of the engine was discussed from the theoretical point of view in Chapter III. The power, of course, will be substantially proportional to the indicated work of one cycle, which is given by equation (12). This work is proportional to the expression $(r^{1.3} - r)/(r - 1)$ the values of which for different values of r are given in a table on page 36.

The subject has been experimentally investigated by Harry Ricardo by means of a variable compression research engine, and his results are given in Fig. 330. Mr. Ricardo found that the indicated m.e.p. varies almost exactly as the air cycle efficiency.

Similar experiments were made on an aircraft engine in France by Paul Dumanois. The engine was an eight-cylinder one of 5.5 by 5.9 in., and originally had a compression ratio of 5.3. By fitting special pistons the compression ratio could be changed to 7.

The results of these tests accorded well with theory. The air-cycle efficiencies corresponding to these two compression ratios are 0.49 and 0.541, the ratio of the latter to the former being 1.10. The engine when run at a speed of 1830 r.p.m. gave 298 and 330 hp. with the two compression ratios, respectively, which shows that the torque increased in the same proportion as the air-cycle efficiency. From these results M. Dumanois drew the conclusion that if it were possible to operate at a compression ratio of 9, the indicated horse power could be increased by 23 per cent as compared with a ratio of 5.3, but owing to greater internal losses the brake horse power probably would be increased only 18 per cent.

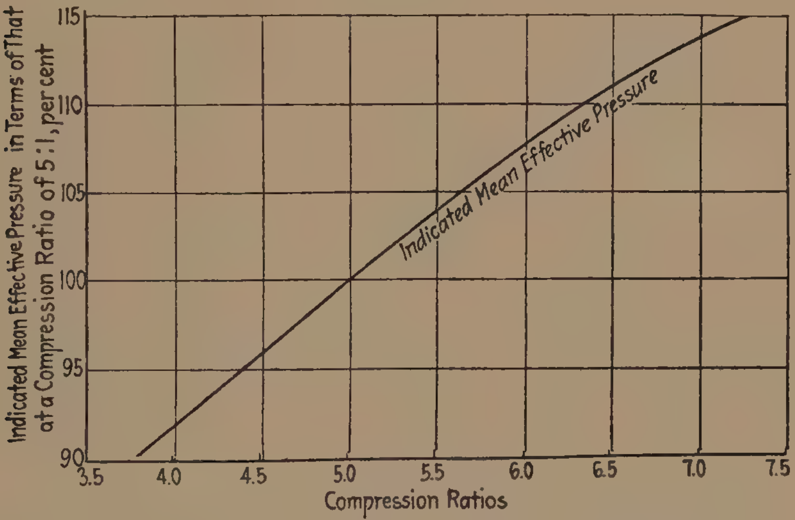


FIG. 330.—VARIATION OF M. E. P. WITH COMPRESSION RATIO.

Among the conclusions drawn from these tests was that although the temperature of combustion increases materially when the compresion ratio is raised, the average temperature of the cycle is not increased, but, rather, decreased, as the theoretical efficiency also increases with the compression ratio. Measurements showed that the piston head and valve temperatures were not affected by the change in compression ratio, and the loss of heat to the cooling water decreased slightly.

Dependence of Power on Atmospheric Pressure and Temperature—In connection with the development of aircraft engines very careful tests have been made to determine the effect on the output of an engine of changes in the barometric pressure and of the atmospheric temperature. At first it was assumed that the output would be directly proportional to the density of the atmosphere, which is directly proportional to the barometric pressure and inversely proportional to the absolute temperature. Very careful measurements made on engines in the altitude chamber of the Bureau of Standards confirmed that the power varies directly as the barometric pressure at the carburetor inlet, but its variation was found to be less rapid than the inverse of the absolute temperature. It was then suggested to allow for this by adding 460° to the standard temperature and the temperature of observation, both in absolute degrees, but further study led to the conclusion that the power varies substantially as the square root of the absolute temperature. That this should be so is a conclusion also arrived at by a consideration of the formula for air flow through a carburetor throat (the derivation of which from first principles is given in Vol. IV of The Gasoline Automobile) viz.,

$$w = \frac{2.04Cpa_1}{\sqrt{T}}\sqrt{\left(\frac{p_1}{p}\right)^{1.42} - \left(\frac{p_1}{p}\right)^{1.71}} \text{ lbs. p. s.,}$$

which shows that the rate of air flow through the carburetor and into the engine is inversely proportional to the square root of the absolute temperature T of the atmosphere.

Therefore, if the standard pressure and temperature be denoted by p and T and the pressure and temperature at which the engine is tested by p_1 and T_1, and if the horse power determined by the test is designated by Hp_1 and the corresponding horse power at standard temperature and pressure (the corrected horse power) by Hp, then

$$Hp = Hp_1(29.92/p_1)(\sqrt{T_1/T}).$$

It is customary to correct the test results to 60° F. The atmospheric pressure is expressed in inches of mercury column, while the temperatures are expressed in absolute degrees.

Bore-Stroke Ratio and Fuel Efficiency—The fallacious idea has been extant that a long stroke results in greater expansion of the burning charge, and consequently in a better utilization of the heat energy in same, but it is directly evident that as long as the compression ratio remains the same the expansion ratio also remains the same. There is, however, a possibility of slightly increasing the thermal efficiency by an increase in the stroke-bore ratio. This is due to the fact that the compression chamber has a more favorable form in the long than in the short stroke cylinder. For instance, if we consider two cylinders of the same displacement, one with a stroke twice the bore and the other with equal bore and stroke; then, if the compression ratio is 4 and the compression chamber is of cylindrical form of the same diameter as the bore, the wall area of the compression chamber in the short stroke cylinder will be about 12 per cent greater than that in the long-stroke cylinder. The advantage with respect to minimum cooling surface is on the side of the long-stroke cylinder over the first half of the stroke, and is particularly pronounced at the beginning of the stroke when the temperature in the cylinder is a maximum.

Bore-Stroke Ratio and Weight Efficiency—There can be little doubt that a long-stroke engine is heavier than a short-stroke, the maximum power outputs being equal. The three heaviest parts of any engine are the cylinders, the crankcase and the flywheel. The weight of the cylinder varies almost as fast as the length of stroke, that of the crankcase varies faster than the length of stroke, and the flywheel weight, for the same coefficient of irregularity, varies as the square of the length of stroke if the mean diameter remained the same, but less rapidly if the diameter is increased with the stroke, as it generally is. Hence the weight of the engine apparently varies at least in direct proportion to the length of stroke while, as previously shown, the output varies less rapidly. This is a relation of considerable importance in aircraft engines, and many of the newer engines of this type have a comparatively small stroke/bore ratio. Thus Packard builds aircraft engines with cylinders having a stroke/bore ratio of only a trifle over 1.00 and the Maybach airship engine cylinders are similarly proportioned.

In most of the more recent passenger-car engines the stroke is approximately 50 per cent greater than the bore, except in the lower priced field, where engines of comparatively low speed are used and where the advantage of the short-stroke engine with respect to weight efficiency seems to be the deciding factor.

Dependence of Fuel Economy on Torque and Speed—When figures of fuel consumption are quoted they are usually the specific consumption at maximum output, or else the specific consumption under conditions of maximum economy. But the specific fuel consumption varies with both the torque and the speed of the engine. Figure 331 shows a diagram prepared by the author for a paper on Unconventional Transmissions presented before the S. A. E. in 1925, and is based on tests made by Herbert Chase on a six-cylinder Pierce-Arrow engine at the Laboratory of the Automobile Club of

America. Engine speeds are plotted on the horizontal and torques on the vertical scale.

The dotted curves in the chart are hyperbolae and represent equal horse powers, while the full lines are equal specific fuel-consumption lines. That is, any point along a full line represents a combination of engine speed and torque for which the fuel consumption per horse power-hour is the same as for any other point along that line, and the specific consumption is marked on the curve.

It will be noted first of all that the specific fuel consumption is much more dependent upon the torque than upon the speed, for all of the equal fuel consumption curves are substantially horizontal. This engine delivers its maximum power at about 1800 r.p.m., and the fuel economy is a maximum at about 1300 r.p.m., substantially throughout the whole torque range. The consumption is slightly higher at the maximum speed and considerably higher at the lowest speeds. For instance, for a torque of 140 lb.-ft. the minimum specific consumption is 0.90 lb. p. hp.-h. at 1350 r.p.m., while at 425 r.p.m. and the same torque it is 1.25 lb. p. hp.-h. It may be said that as a rule the fuel economy increases with the speed up to about two-thirds the speed of maximum output, and decreases with further increases in speed up to the maximum.

How the fuel economy decreases with the torque may also be clearly seen. The variation of the specific fuel consumption with the torque load at the speed of 1350 r.p.m. is given in the following table:

Torque Load (lb.-ft.)	270	190	140	60	12
Fuel (lb. p. hp.-h.)	0.70	0.80	0.90	1.25	2.00

Reasons for Variation in Fuel Economy—That the specific fuel consumption increases rapidly as the torque decreases is accounted for largely by the fact that some of the mechanical losses are independent of the torque while others actually increase as the torque decreases. For instance, the bearing friction is substantially independent of the torque, while the pumping losses may increase when the throttle is closed to reduce the torque, as that results in a greater vacuum in the cylinders during the inlet stroke and, consequently, a greater gaseous pressure for the piston to work against. As a rule, however, this increase in pumping loss during the inlet stroke is about balanced by a decrease in the pumping loss during the exhaust stroke; but even with a constant internal loss it is obvious that the economy must decrease as the torque load decreases.

It was stated above that with increase in speed the fuel efficiency at first increases, while from about two-thirds the speed of maximum output on it decreases. This may be explained as follows: The heat energy of the fuel that is being liberated by the combustion is divided into four parts. One part is converted into mechanical energy; a second part goes into the water jacket, a third into the engine frame (from which it is dissipated to the adjacent atmosphere), and the fourth part passes off with the exhaust. Now, if the engine speed is very low, the gases remain in contact with the cooling walls for a relatively long time, and more of the heat will be lost to the engine frame and jacket. With increasing engine speed these losses are cut down, and if these were the only factors affecting the fuel efficiency we might expect a continual decrease in specific fuel consumption up to the highest speeds.

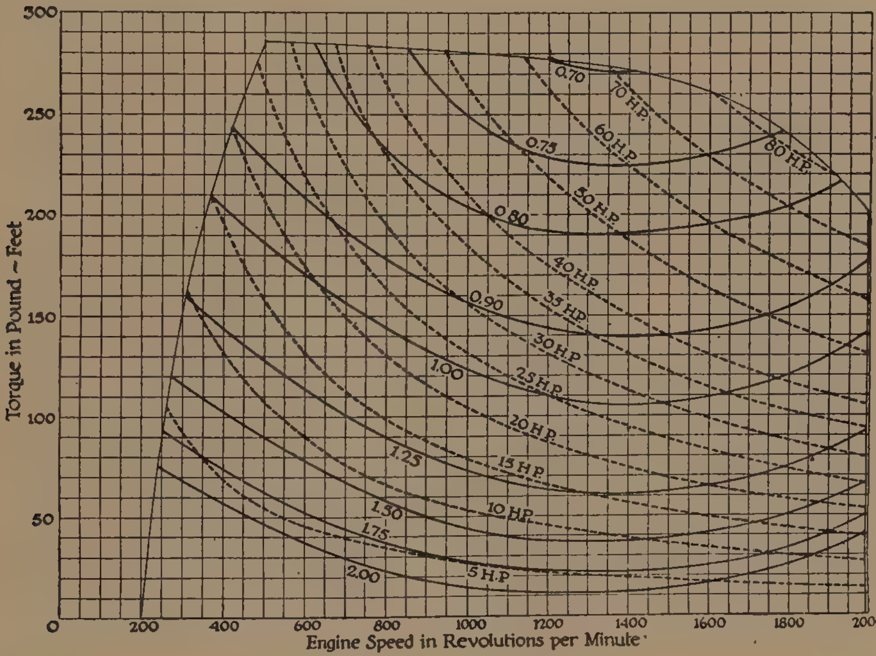


Fig. 331.—Variation of Specific Fuel Consumption with Speed and Torque.

However, the specific fuel consumption is based on the brake horse power-hours, that is, upon the difference of the heat energy actually converted into mechanical energy and the internal losses, and it was shown in the foregoing that the mechanical losses in terms of horse powers increase quite rapidly with the speed, a fraction increasing directly as the speed and the remainder as the square of the speed. This, of course, tends to reduce the fuel economy as the speed is increased. At very low speeds the other influence—variation of jacket and radiation loss with speed—is more potent, but from about two-thirds the speed of maximum output on, the growing mechanical losses dominate, and the fuel efficiency therefore decreases.

Variation of Fuel Economy with Mixture Ratio—It was shown in Chapter I that the theoretical combining ratio of air and petroleum fuel is about 15.2 : 1, but it was also pointed out that mixtures both far richer and considerably leaner are ignitable. The driver, of course, has no way of determining the mixture proportion, except in so far as a change in the proportion affects the power of his engine. Under most conditions, of course, it is desirable to use a mixture proportion which results in the highest thermal efficiency (lowest fuel consumption per horse power-hour). It is therefore well to know how the engine power and economy vary with the mixture ratio.

This problem was investigated for the National Advisory Committee for Aeronautics by Stanwood B. Sparrow of the Bureau of Standards. Mr. Sparrow constructed the curve, Fig. 332, which shows the varying relationship between indicated mean effective pressure and specific fuel consumption. It should be pointed out that the values of the i.m.e.p. given in the chart do not apply to automobile engines, as most of the tests on which the results were based were made in the Altitude Chamber of the Bureau of Standards and have reference to operation of the engine in a rarefied atmosphere. The variation of the thermal efficiency with the mixture ratio would be similar, however, in a denser atmosphere, that is, in an automobile engine.

Starting with the leanest possible mixture, we obtain the results represented by the left-hand end of the curve. By increasing the mixture strength the specific fuel consumption is cut down and the i.m.e.p. materially increased. This shows that the leanest possible mixture is not the most economical one.

The lowest point of the curve represents the most economical mixture. By continuing to increase the strength or richness of the mixture the power of the engine is further increased, as the i.m.e.p. increases, but the fuel consumption also increases. The point of the curve farthest to the right represents the most powerful mixture.

Mr. Sparrow found that when using gasoline as fuel, maximum power usually is obtained with air-fuel mixtures of between 14.3 : 1 and 12.4 : 1. Nearly minimum specific fuel consumption results from decreasing the fuel content of the charge until the power is less than the maximum value by 5 per cent. Thus not very much can be gained in the way of power by enriching the mixture, while much can be lost in the way of fuel economy by using too rich a mixture. The curve shows, for instance, that the same i.m.e.p. is obtained with a lean mixture resulting in a specific consumption of only 0.45 lb. per i.hp.-hr. and a rich mixture resulting in a specific consumption of 0.8 lb. per hp.-hr.

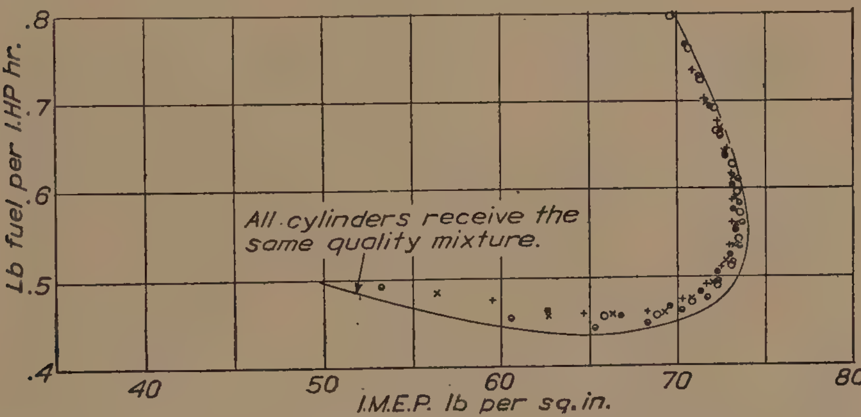


FIG. 332.—VARIATION OF BRAKE M. E. P. WITH SPECIFIC FUEL CONSUMPTION WHEN MIXTURE RATIO IS VARIED.

Effect of Compression on Fuel Economy—What is generally referred to in thermo-dynamic discussions as the air-cycle efficiency is determined by means of the equation

$$e = 1 - \frac{1}{r^{\gamma - 1}},$$

in which γ is taken at its full theoretical value, viz. 1.405, or 1.4. Although the actual thermal efficiency is much lower than that calculated from this equation, it has been found that the proportional change in the actual thermal efficiency resulting from a change in the compression ratio, is substantially the same as the proportional change in the air-cycle efficiency.

An analysis of the saving in fuel which would be made possible by an increase in the compression ratio of an automobile engine from 4.5 to 6.5 has been made by Stanwood W. Sparrow of the Bureau of Standards. Since the air-cycle efficiency for a 6.5 compression ratio is 17 per cent higher than that for a 4.5 ratio, the fuel consumption per indicated brake horse power-hour would be decreased in the ratio of 117 to 100. There is, however, an additional gain, since for the same maximum power the engine with the higher compression ratio can be made smaller, and the friction loss reduced in consequence.

Assuming friction losses to be proportional to displacement, the mechanical efficiency would be increased from 90 to 91.5 per cent by this reduction in displacement, or from 80 to 82.9 per cent, and it was shown by Mr. Sparrow that for full-load operation the increase in mileage from a gallon of gasoline would be between 18 and 19 per cent if the mechanical efficiency of the original engine had been 90 per cent and between 20 and 21 per cent if the original mechanical efficiency had been 80 per cent. At part load the gain in mileage would be even greater, and Mr. Sparrow found that at one-half full power the gain due to a change in compression ratio from 4.5 to 6.5 would amount to 27 per cent.

Heat Balance—The heat energy of the fuel burned in an engine is in small part converted into mechanical energy, the remainder being discharged through three different channels: A large portion passes into the jacket water and is carried by it to the radiator; another portion is carried off with the exhaust gases, and the remainder is dissipated from all the exposed surfaces of the engine and adjacent parts to the surrounding atmosphere. A statement of the proportions of the total heat supplied entering each of these four channels—mechanical energy, jacket loss, exhaust loss and general radiation—under any particular set of operating conditions is known as a heat balance.

If the amounts of energy represented by each of the different items are measured separately and then added up, it is found that the sum is not quite equal to the amount of heat contained in the fuel burned, and the difference is put down as "heat unaccounted for." That some heat is not accounted for is partly at least due to the fact that combustion is never quite complete and some unburned fuel passes off with the exhaust gases. The mechanical energy is measured by means of a dynamometer. The heat entering the cooling water is determined by measuring the temperatures of the water on entering and leaving the jacket and the rate of water flow. The heat in the exhaust gases is determined by measuring the temperature of the exhaust gas by means of a thermocouple close to the engine, making an analysis of the exhaust gases, calculating the specific heat of same from the composition found, and determining the rate at which exhaust gases are discharged by measuring the air and fuel flows. The heat lost by radiation can be measured by enclosing the engine in an air hood resembling the bonnet of the car and connecting same by an air-tight chute to a blower. Then, by measuring the rate of air flow through the hood and the rise in temperature in passing through it, the loss can be calculated by taking account of the specific heat of air.

A series of tests for the purpose of establishing heat balances for the entire speed and torque range of a six-cylinder passenger-car engine were run at the Laboratory of the University of Michigan in 1916 and reported in an S. A. E. paper by Profs. Fishleigh and Lay. The tests were made on an L-head $4\frac{1}{8}$ by $5\frac{1}{4}$ in. engine with splash lubrication. Runs were made at 640, 1000 and 1350 r.p.m. and at power outputs varying by steps of 5 hp., starting with, 5, 10 and 15 hp. at the three speeds respectively.

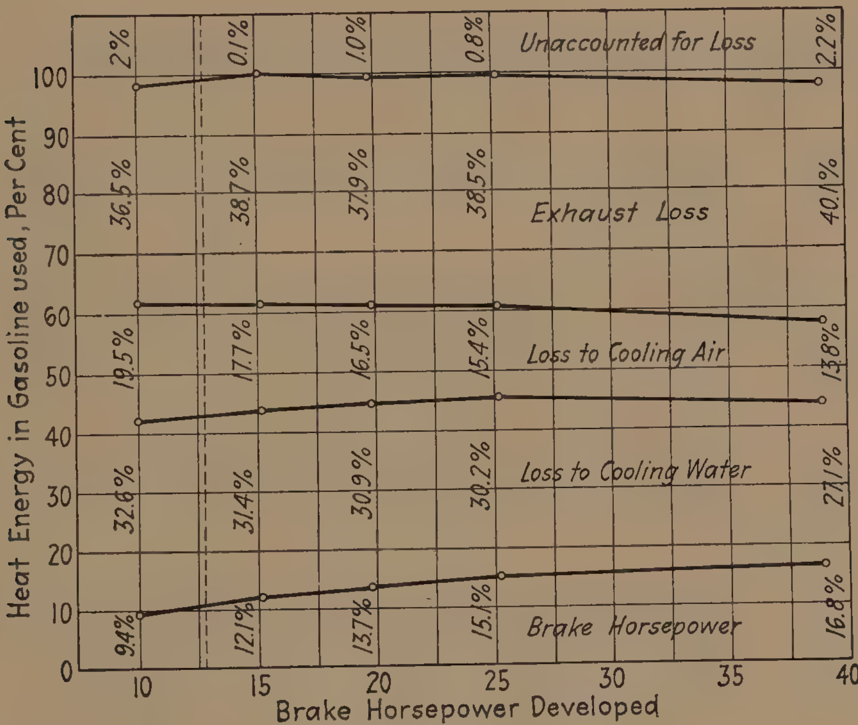


FIG. 333.—HEAT BALANCE OF A SIX-CYLINDER ENGINE AT 1000 R.P.M.

The heat balance sheet for 1000 r.p.m. is reproduced herewith (Fig. 333), and illustrates the method of plotting such sheets. The thermal efficiency of the engine in this series of runs was not very high, which is believed to have been due to the fact that the carburetor was adjusted to give a rather rich mixture. For instance, at 20 hp. the fuel consumption was at the rate of 0.93 lb. per hp.-hr. In these runs the thermal efficiency increased from a trifle under 10 per cent at 10 hp. to 15 per cent at 25 hp. to 17 per cent at 38 hp.; the mechanical efficiency increased from 53 per cent at 10 hp. to 76 per cent at 25 hp. to 86 per cent at 38 hp., while the fuel consumption per brake hp. decreased from 1.07 lbs. per hp.-hr. at 10 hp. to 0.84 lb. at 25 hp., to 0.76 at 38 hp.

It is of interest to quote here some heat balance test results on high-compression aircraft engines obtained in the Altitude Chamber of the Bureau of Standards. In the following table:

E J B is the sum of the heats accounted for in the exhaust, jacket and brake, expressed as per cent of the heat supplied in gasoline.

E is the heat rejected in exhaust, expressed as per cent of heat accounted for.

J is the heat removed in jacket water, expressed as per cent of heat accounted for.

B is the brake or dynamometer horse power, expressed as per cent of heat accounted for.

Table XII—Heat Balances of Aircraft Engines at Ground Level

Comp. Ratio	4.7	5.3	5.4	7.2
E J B	91.5	85.4	85.5	85.6
E	50.3	55.9	55.1	50.4
J	23.9	16.4	17.3	18.7
B	25.8	27.8	27.6	31.0
Engine	Hisp.-150	Hisp.-300	Liberty	Liberty
R.P.M.	1500	1600	1600	1600

In order to find the brake thermal efficiency on the basis of heat contained in the fuel consumed the figure in the B line must be multiplied by the percentage in the E J B line. This gives for the Liberty engine with 7.2 compression ratio on the special Hecter fuel a brake thermal efficiency of 26.5 per cent.

Friction Horse Power—The difference between the indicated and the brake horse power of an engine is known as the friction horse power. The right way to determine this friction horse power would seem to be to take an indicator card at full load, determining from it the indicated horse power, making simultaneously a brake test to ascertain the brake horse power, and then taking the difference. However, the determination of the indicated horse power by means of an indicator involves considerable difficulty, and there are many possible sources of error. Therefore, the friction horse power is generally determined by ''motoring'' the engine, that is, driving it by means of an electric motor (or dynamometer), and by measuring the voltage and current consumption of the motor and from its efficiency curve, it is then possible to determine the power required to turn over the engine at various speeds.

The internal losses in an engine are essentially of two kinds, viz., pumping losses and frictional losses. During the inlet and exhaust stroke the gaseous pressure on the piston is greater on its forward side (on the under side during the inlet and on the upper side during the exhaust stroke), hence during both strokes the piston must be moved against a pressure, and this occasions the so-called pumping loss. The frictional loss is made up of the friction between the pistons and cylinder walls, piston rings and cylinder walls, and between the crankshaft and camshaft and their bearings, as well as by the loss occasioned by driving the essential accessories, such as the water pump, ignition unit, etc. Fig. 333-A, taken from an S. A. E. paper by S. W. Sparrow, shows the effect of speed and inlet depression on the friction loss.

The internal losses at any speed may be expressed either in horse powers, in terms of friction torque or in terms of a loss in mean effective pressure (in pounds per square inch of cylinder head area). Numerous tests have shown that the friction torque is made up of one part independent of the speed and another which varies directly with the speed. Remembering that horse power is equal to torque times speed, it follows that the friction horse power includes one item varying in direct proportion to the speed and another varying as the square of the speed.

The pumping loss at any speed depends largely upon the relative size of the valves, while the friction losses depend upon the character and fit of the bearing surfaces, the lubricant used, etc. It is therefore obvious that the relative magnitudes of the two classes of loss must vary considerably in different engines. In a lecture given at Kings College, London, in 1925, Harry Ricardo stated that the overall mechanical efficiency of an internal combustion engine of the automotive type at full load is generally between 80 and 90 per cent. However, an 80 per cent efficiency at full load drops to 55 per cent when the torque load is only 30 per cent of the maximum, as it often is in the operation of automobiles, and a 90 per cent efficiency drops to 73 per cent. He further stated that of the gross mechanical losses at full load, piston friction accounts for about 60 per cent, pumping loss for about 25, and bearing friction and the driving of accessories, for 15 per cent. A plot of friction torque against engine speed is substantially a straight line, while a plot of friction horse power against engine speed is a curve convex toward the horizontal or speed axis. The loss in mean effective pressure due to internal losses—which, of course, is the difference between the indicated and brake mean effective pressures—varies from 7 or 8 lbs. per sq. in at the lowest idling speed, to 20 or 22 lbs. at the speed of maximum output.

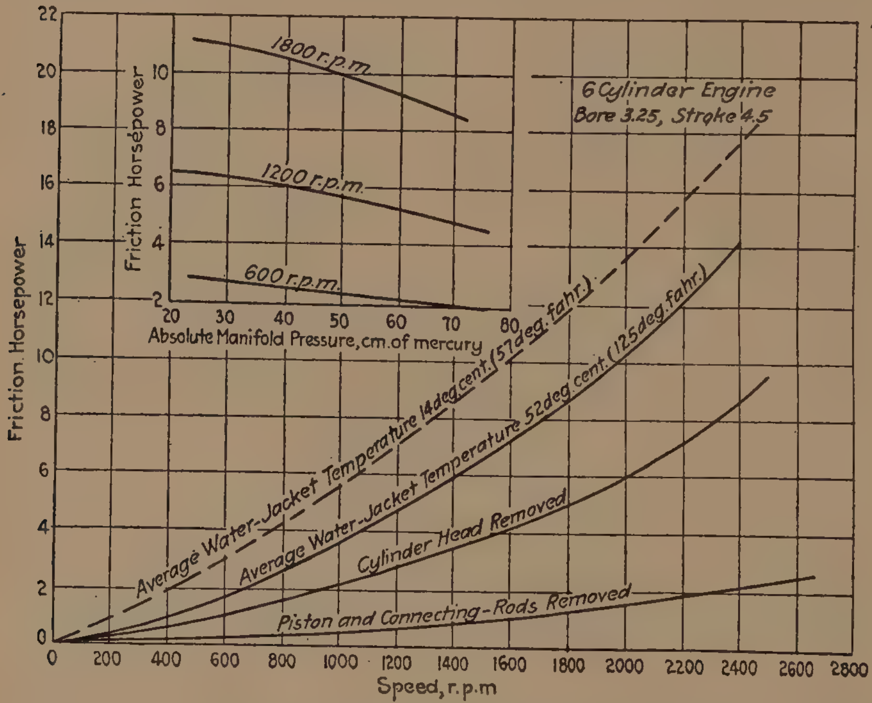


FIG. 333-A.—RESULTS OF FRICTION HORSE POWER TESTS ON A PASSENGER CAR ENGINE.

Characteristic Curves—Curves which show the variation of the torque, brake mean effective pressure, horse power, friction horse power, friction torque or specific fuel consumption with the speed are known as characteristic curves. If such curves are plotted for a particular engine the speed in r.p.m. is usually plotted on the horizontal axis and the horse power, torque etc., are plotted on the vertical axis. If it is desired to compare engines of different length of stroke by means of such curves it is a better plan to express the speed in terms of feet per minute piston speed, rather than in r.p.m.

In Fig. 334 are shown piston speed—b.m.e.p. curves of what may be regarded as the extreme types of passenger-car engine—the low-speed type with a stroke of about 4 in. and delivering its maximum power at a speed of about 1500 or 1600 r.p.m., and the high-speed type of 5¼ or 5½ in. stroke, and delivering its maximum power at from 3200 to 3400 r.p.m. The piston speeds of maximum output are roughly 1000 and 3000 ft. p.m., respectively. There is also a wide difference in the brake mean effective pressures of these engines. In the low-speed engine the b.m.e.p. has a maximum value of about 75 lbs. per sq. in. and decreases to about 55 lbs. per sq. in. at the speed of maximum output. In the high-speed engine, on the other hand, the b.m.e.p. attains a maximum value of nearly 120 lbs. per sq. in. and is only a shade below 100 lbs. per sq. in. at its speed of maximum output. Since the b.m.e.p. is nearly twice and the piston speed nearly three times as great, the high-speed engine develops almost six times the hp. per unit of displacement that the low-speed engine does.

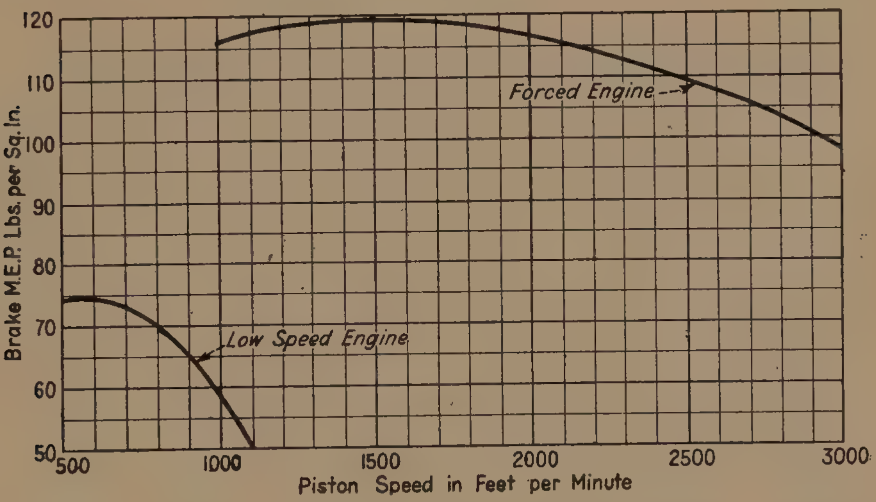


FIG. 334.—M. E. P.-PISTON SPEED CURVES FOR EXTREME PASSENGER CAR ENGINE TYPES.

As pointed out, these two curves are characteristic of extremes, and the performance of practically all passenger-car engines lies somewhere between these limits. The curves shown in Fig. 335 are characteristic of most of the newer

American passenger-car engines. The maximum b.m.e.p. is slightly below 100 lbs. per sq. in. and the b.m.e.p. at the speed of maximum output about 80 lbs. per sq. in., this speed being equivalent to about 2500 ft. p.m. piston speed.

In the same figure is shown a horse-power curve for a six-cylinder 3 by 5 in. engine whose b.m.e.p. varies as shown. The b.m.e.p. curve also serves as torque curve for this engine, for with a given piston displacement the torque at any speed is directly proportional to the b.m.e.p.

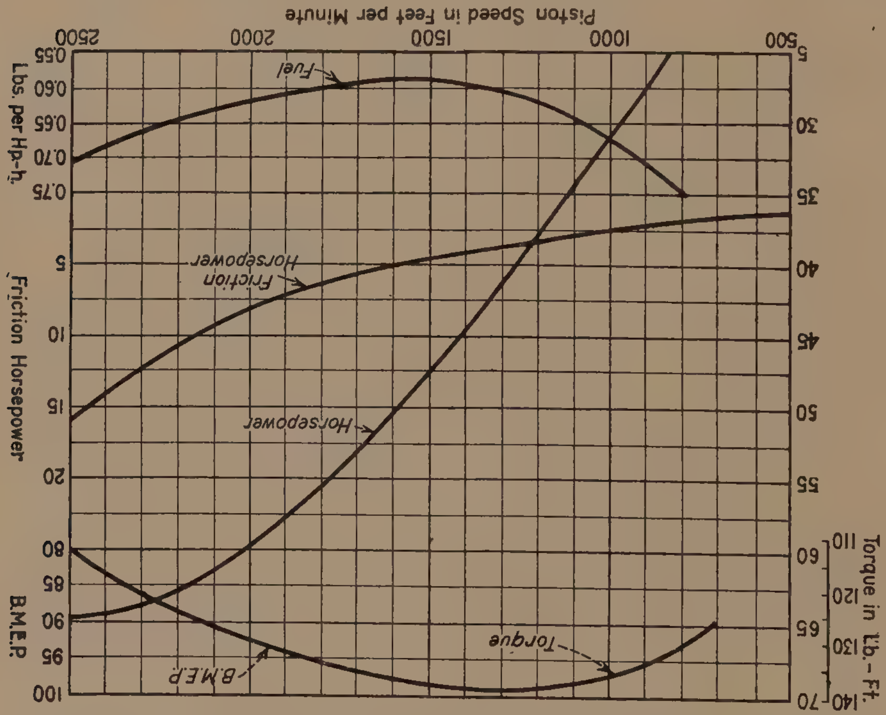


FIG. 335.—CHARACTERISTIC CURVES OF A MODERN SIX-CYLINDER 3 BY 5-IN. PASSENGER-CAR ENGINE.

On the same chart are also plotted curves of specific fuel consumption and friction horse power for the six-cylinder 3 × 5 in. engine. At the speed of maximum output the friction horse power is exactly one quarter the brake horse power, hence the mechanical efficiency is 80 per cent. Since at this speed the brake horse power begins to decrease and the friction horse power increases at a high rate, the mechanical efficiency varies rapidly. The lowest fuel consumption usually corresponds to a speed equal to or slightly higher than the speed of maximum torque, and the economy of operation becomes lower toward both ends of the speed scale. The minimum specific fuel consumption here shown is about 0.58 lb. per hp.-hr., which is a very good performance.

Inlet Pressures—During the inlet stroke there is an under pressure or partial vacuum in the cylinder and also in the inlet manifold. This inlet vacuum is used for a number of purposes in connection with auxiliary apparatus on automobiles, as for raising the fuel from a main tank suspended from the car frame at the rear to a vacuum tank on the cowl board, for operating a windshield cleaner and even for applying the brakes. The vacuum, of course, varies with the speed of the engine, and with the throttle position, increasing with the former and as the throttle approaches the closed position.

Whenever the rate of air flow through the inlet passage is high, the vacuum in the cylinder will be appreciably higher than that in the manifold, because of the considerable pressure drop caused by the resistance of the valve under that condition; but at low engine speeds and when the throttle valve is nearly closed, there is no appreciable difference between the two values.

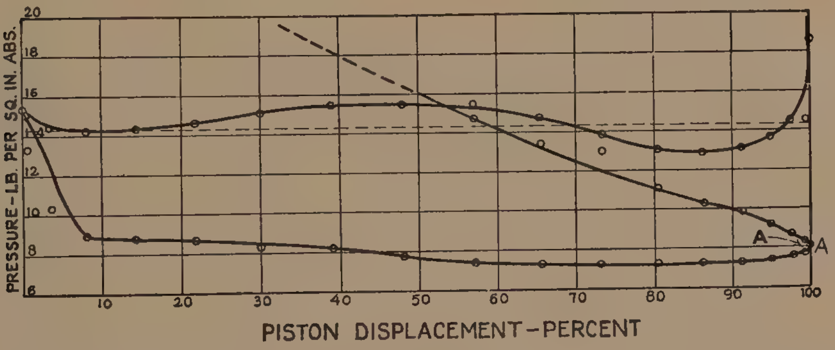


FIG. 336.—CYLINDER PRESSURES DURING EXHAUST, INLET AND PART OF COMPRESSION STROKES WITH ENGINE THROTTLED TO 8 LBS. PER SQ. IN. INITIAL PRESSURE.

A series of tests to determine cylinder inlet pressures were made at the Engineering Experiment Station of the University of Illinois on a Ford engine. Under full throttle at low speed (800 r.p.m.) the depression in the cylinder was 0.8 lb. per sq. in. On the other hand, in tests made on a Pierce Arrow engine at the Laboratory of the Automobile Club of America a minimum depression on the inlet manifold directly above the carburetor of 0.5 in. of mercury column was measured, which is equal to about 0.25 lb. per sq. in. From these minimum values at low speed the depressions increased until at the speed of maximum power they became about 2.7 lbs. per sq. in. in the cylinder in the one case and 1.8 in. of mercury or nearly 1 lb. per sq. in. in the manifold in the other. This clearly shows the effect of the inlet valve in keeping down the vacuum in the manifold below that in the cylinder at high air velocities. With decreasing throttle opening the vacuum in both the cylinder and the manifold increases. The maximum vacuum measured in the cylinder of the Ford engine was 9.2 lbs. per sq. in. at 20 per cent throttle opening and 1000 r.p.m., while the lowest measured in the manifold of the Pierce-Arrow engine was 18.1 in. of mercury column or 9.5 lbs. per sq. in. The figures of pressure here given are under pressures, that is, the difference between atmospheric pressure and the absolute pressure in the cylinder or manifold.

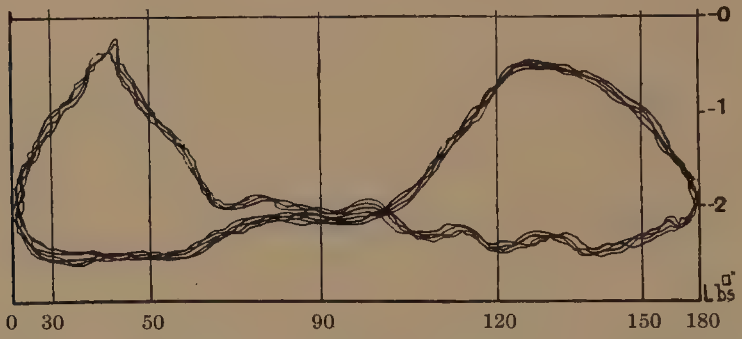


Fig. 336-A.—Pressures in Inlet Manifold of Four-Cylinder Engine Under Full-Load Conditions.

Figure 336 shows curves of exhaust and inlet pressure in the cylinder of an engine running at about one-half its normal maximum speed and with the throttle only very slightly open, so that at the beginning of the compression stroke there is a vacuum of 8 lbs. per sq. in. in the cylinder. The cylinder naturally draws in very little charge under these conditions and there is therefore practically no back pressure on the piston during the exhaust stroke. Fig. 336-A shows the pressure variations in the inlet manifold of a four-cylinder engine under full load at normal speed, during the period of two strokes. In this case the horizontal scale measures time or crank motion rather than piston motion. At the very beginning of the inlet stroke there is quite a vacuum in the manifold, which is due to the preceding cylinder, whose inlet valve remains open for 30° or more after the dead center position (0°). It will be observed that about 50° from the beginning of the stroke the vacuum in the manifold almost vanishes. This is due to the fact that the inlet valves of both the cylinder whose inlet period is coming to an end and that whose inlet period is just beginning, have been for some time either closed or nearly closed, and the manifold therefore has had time to fill up with charge from the carburetor.

Supercharging—From the very beginning of the automobile industry, constant efforts have been made to increase the power obtainable from an engine of given displacement, and one of the favorite expedients used to this end has been that of supercharging. This process consists in forcing into the cylinders an amount of charge greater than can be drawn in by the pumping action of the working pistons. The use of superchargers for automotive purposes was first suggested in connection with racing engines, but the real practical development of the process dates from the latter years of the War, when preparations were being made to use it on aircraft engines at great altitudes.

Normally the power of aircraft engines falls off with increase in altitude, because of the decrease in the density of the air. At an altitude of 18,000 ft., for instance, an engine draws in only about half as much charge per cycle as at sea level, and therefore can develop only half as much power. By means of an air pump or a supercharger it is possible to force into the cylinders at this altitude as much charge as they normally receive at sea level. In this way it is possible to keep the power of aircraft engines constant up to altitudes of 20,000 ft. and more.

In recent years superchargers have been much used on racing engines. Racing rules place a limit on the displacement of the engines; the use of a supercharger enables the engine to develop a greater output, which in turn adds to the speed of the car. When a cylinder is supercharged, the pressure within it is greater than it would otherwise be, throughout the power stroke.

Types of Supercharger—Two types of supercharger or charge pumps have been used, the centrifugal blower and the Root's type of blower. These are illustrated in Figs. 337 and 338, respectively. The centrifugal blower consists of an impeller fitting closely inside a housing in which it is rotated at speeds up to 30,000 r.p.m. In the case of superchargers for 122 cu. in. engines, the diameter of the impeller is about 7½ in. Impellers are made either of alloy steel, duralumin or magnesium. At such tremendous speeds the centrifugal and inertia forces on the impeller naturally are very great, and the material whose tensile strength—weight ratio is the highest is the most advantageous. The housing is generally made of cast aluminum.

Just outside the space which is swept by the impeller there is sometimes a diffuser, which is designed to so direct the flow of air that its kinetic energy is converted into static energy with as little loss as possible. Outside of this diffuser, which is merely an annular space containing curved blades or vanes, there is a volute which receives the air set in motion by the impeller and conducts it to the outlet of the housing.

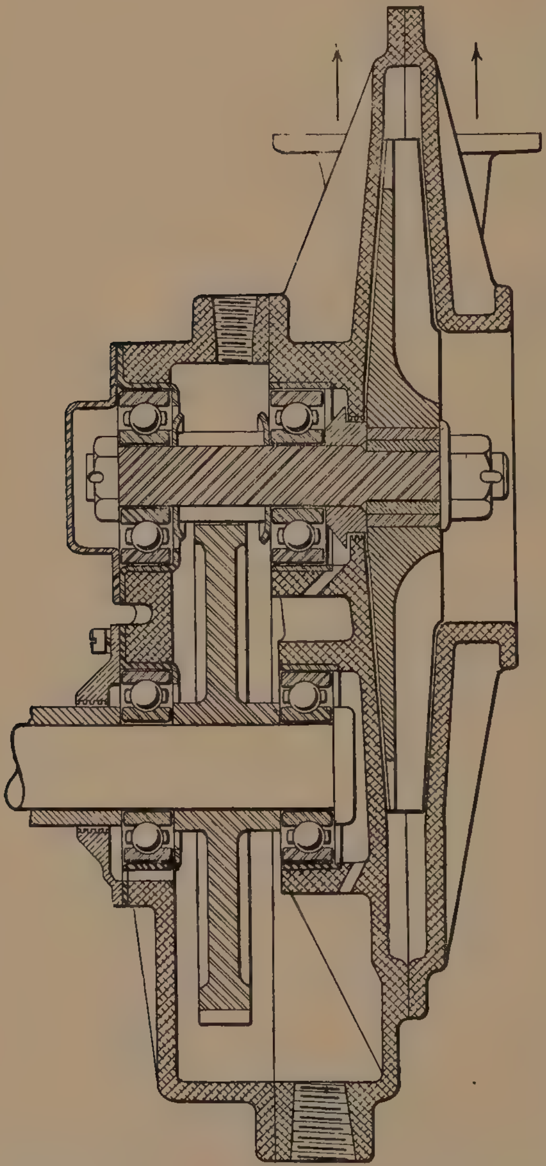


Fig. 337.—Centrifugal Blower.

The Root's blower type of pump or supercharger consists of two rotors of a form similar to that of a figure eight. These are geared together and rotated at the same speed, and this type of blower is related to a gear type of pump in its principle of operation.

The characteristics of the two types of pump are quite different. With the Root's blower type the rate at which air is moved varies almost if not quite as fast as the speed of rotation, if the air is moved against a constant resistance. With the centrifugal blower type, on the other hand, it varies substantially as the square of the speed. With a Root's blower type of pump connected to an engine and driven from it at constant speed, the degree of supercharge is therefore almost constant, being slightly greater at low than at high speeds. With a centrifugal blower, on the other hand, the supercharging effect is small at low speeds.

Installation of Supercharger—There are two methods of arranging the supercharger relative to the carburetor and engine. It may be placed either ahead of the carburetor or between the carburetor and the engine. In most of the recent racing cars fitted with supercharger, the latter was placed between the carburetor and the engine. One advantage of this is that then the fuel can be fed to the carburetor by a gravity system, whereas if the supercharger is located ahead of the carburetor the fuel must be fed under pressure. A disadvantage of this arrangement is that the supercharger handles explosive mixture instead of pure air, and in case of a back-fire through the carburetor there will be an explosion in the supercharger and in the inlet tract between it and the cylinders. In the case of aircraft engines, the supercharger and inlet manifold of which are of large volume, a relief valve is usually provided on the inlet manifold, so that in case of an explosion in the inlet system no excessive pressure can develop. This is merely a spring-seated poppet valve in the inlet manifold wall opening outwardly under pressure. In racing-car engines the volume of the super-

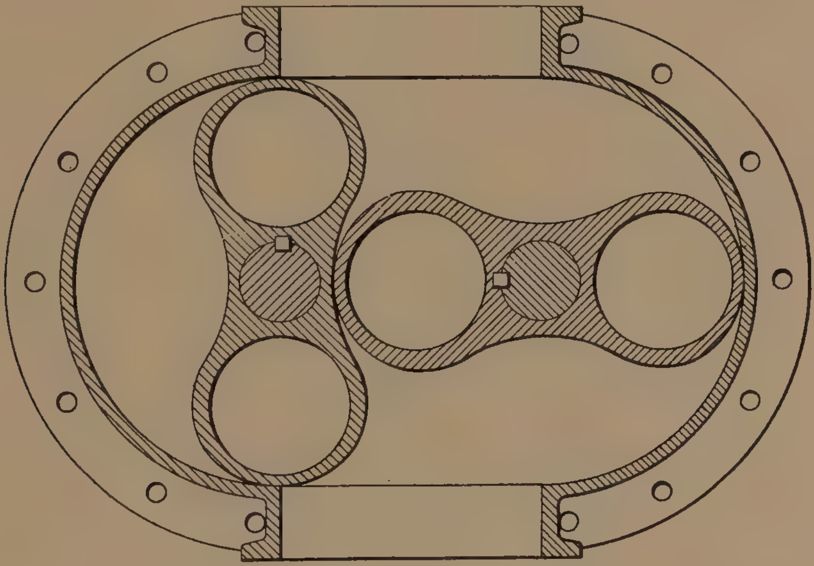


FIG. 338.—ROOTS' TYPE OF BLOWER.

charger and the inlet manifold is very much smaller and the wall thicknesses are relatively heavier, hence a relief valve is considered unnecessary. However, in order to reduce the volume of combustible charge contained in the system to a minimum, the diffuser is sometimes omitted from the compressor, which, of course, reduces the efficiency of conversion of kinetic to static energy somewhat. With the carburetor between the supercharger and the engine the float chamber of the carburetor must be placed in communication with the delivery side of the pump. The two methods of installation are illustrated in Fig. 339.

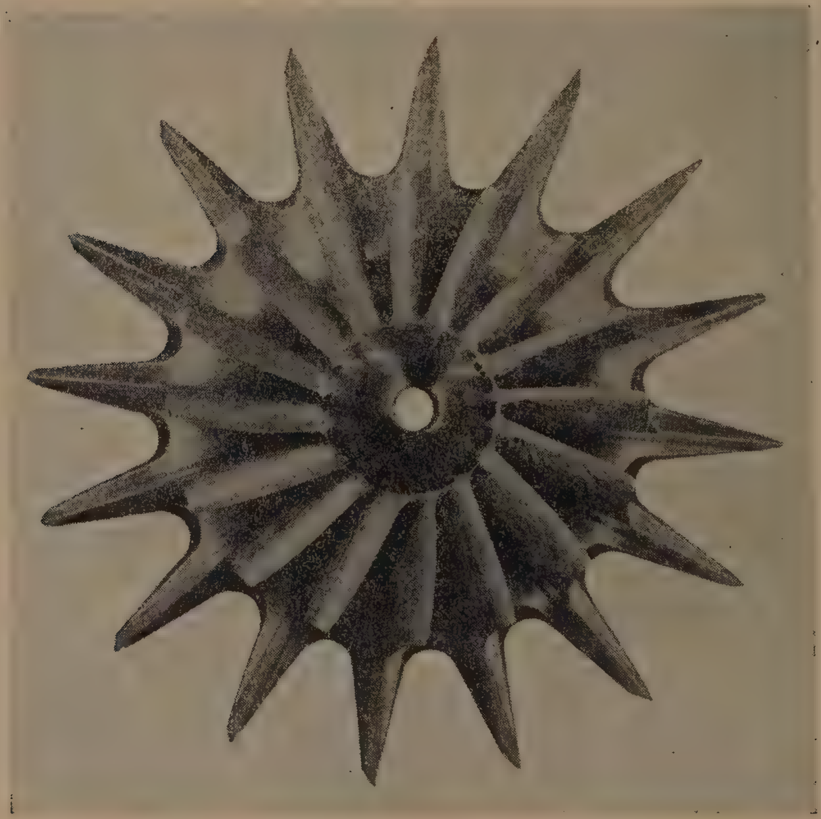


FIG. 338-A.—MAGNESIUM IMPELLER OF SUPERCHARGER.

Some Effects of Supercharging—One advantage of a supercharger is that the compression of the air is accompanied by the generation of heat; this heat raises the temperature of the air and the warmer air facilitates the vaporization of the fuel, hence the distribution is generally better in a supercharged engine. Another advantage resulting from the use of a supercharger is that, since the charge flows through the cylinder valves under considerable pressure, it is not necessary to make the valves as large and to lift and seat them as rapidly as is otherwise necessary in a high-speed engine. Whereas formerly there was always a good deal of valve trouble with racing engines, since the introduction of the supercharger this has practically vanished.

Mercedes Supercharger Engine—Practically the only stock car using a supercharger at the present time is the Mercedes Kompressor, which employs a Root's blower type. This is driven from the crankshaft through a friction clutch controlled through a connection to the accelerator pedal. Normally the clutch is disengaged and the supercharger is not working, but if the driver depresses the accelerator pedal all the way, that is, when he wants more power than the engine is capable of developing without supercharging, the clutch is let in and the supercharger starts to work.

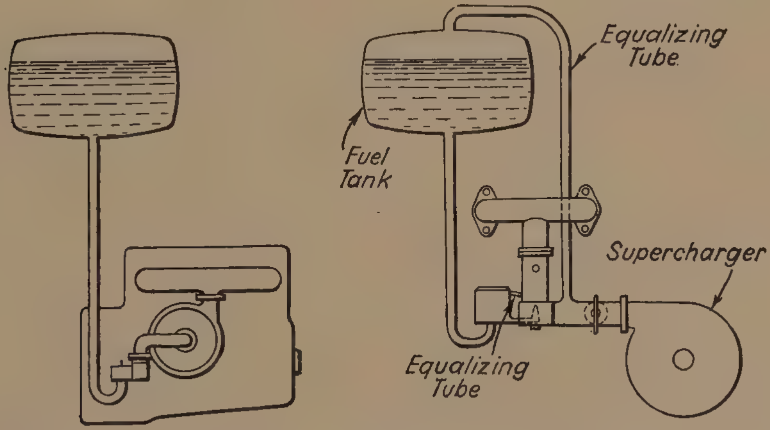


FIG. 339.—TWO METHODS OF ARRANGING A SUPERCHARGING SYSTEM.

In European countries where there is a high annual tax on automobiles calculated on the basis of piston displacement, the use of a supercharger offers a certain advantage in that it permits of fitting a car with an engine of small displacement and still having high maximum speed and high acceleration. There is, moreover, an advantage from the standpoint of fuel economy, for the smaller engine operates at a higher average load factor than a larger one would, and this favors fuel economy.

The six-cylinder Mercedes has a bore of 3.15 and a stroke of 5.12 in. and is designed for a normal compression ratio of 5.4, which is permissible with benzol blends used in Germany as fuel. Without supercharger the engine develops a maximum of about 60 hp. while with supercharger the maximum hp. is about 100 at 3000 r.p.m. The volumetric efficiency at 3000 r.p.m. is said to be about 100 per cent. At this speed about 10 hp. is required to drive the supercharger. On racing engines the cylinders sometimes are supercharged to 10 lbs. above atmosphere.

The Engine as a Brake—When the spark is cut off and one of the lower gears engaged, the gasoline engine acts as an excellent vehicle brake. Its braking effect is due chiefly to mechanical friction, but also to a certain extent to so-called pumping losses, although the gas compressed in the cylinder during the compression stroke expands again during the following stroke and returns a large part of its stored energy to the moving parts.

The braking effect could be considerably increased by opening the exhaust valve at the beginning of the third stroke and holding it open during the whole of that stroke. The charge compressed during the second stroke would then escape through the exhaust valve and its energy would not be returned to the moving system. This, however, requires shifting of the camshaft and involves complications which American engineers do not consider worth while. An experimental investigation of engine braking was made by Dr. W. Watson in England a good many years ago. It was found that the braking effect obtainable with the ordinary engine with the throttle closed is equal to 33 per cent of the full-load torque of the engine; with the throttle closed and the compression cock open it increases to 46 per cent; with the exhaust valve prevented from opening it is 43 per cent and with the exhaust valve opened during the third stroke it is equal to 65 per cent of the full-load engine torque.

Saurer Engine Brake—Engine braking is used on several European trucks, the best known among these being the Saurer made in Arbon, Switzerland. This engine is of the L-head type and its single camshaft carries its driving gear on a steep-pitch worm thread. The cams are of considerably greater width than the roller followers, which permits of shifting the camshaft endwise by means of the throttle lever on the steering wheel, by continued motion after the throttle is fully closed. In this way the camshaft is turned 75° around its axis, the opening period of the inlet and exhaust valves are advanced 150° of crank motion, and a third, auxiliary cam is brought into action, opening the inlet valve during the latter part of what is normally the inlet stroke. When the cam gear is set for braking the action is as follows:

The regular inlet valve closes soon after the beginning of the inlet stroke, but it is opened again a moment later by the auxiliary cam (forming a unit with the regular inlet cam), and closes at about the same time as in normal operation. During the following up-stroke the air is compressed in the cylinder. At about the end of this stroke the exhaust valve (because of its 150° extra advance) begins to open and lets the compressed air escape into the atmosphere. It is the compressing of the air in the cylinder during the second stroke that is responsible for most of the braking effect. The exhaust valve remains open during the third stroke, and when it closes the inlet valve opens and remains open during the fourth stroke, which is an up-stroke. At this time there is an air pressure of about 20 lbs. in the inlet manifold.

Weight of Engines—Passenger-car and truck engines with cast-iron cylinders and crankcases weigh very close to 2.5 lbs. per cu. in. of piston displacement. With aluminum crankcases and larger bores the weight reduces to 2 lbs. per cu. in. displacement and even a little less. This applies to the weight of the engine without carburetor and ignition unit. On a displacement basis the weight of engines has been materially increased during the past 15 years. For instance, in the earlier editions of this book rules were given for the weight of engines with aluminum crankcases which made the weight per cubic inch displacement 1.59 lbs. for a four-cylinder and 1.49 lbs. for a six-cylinder. This is partly due to the fact that smaller cylinder bores are now used and it is impossible to decrease wall thicknesses in direct proportion to the bore, and partly to the fact that crankshafts and crankcases are made much heavier. At the same time, however, the power output from a given displacement has been greatly increased and the weight per horse power is now much less. For engines running at about 2000 ft. p.m. piston speed the average weight is about 10 lbs. per hp., for truck engines running at 1000 to 1200 ft. piston speed weight it is around 20 lbs. per hp.

In order to reduce the weight per horse power, the first essential is that the reciprocating parts be made as light as possible, so as to permit of high piston speed and consequently increased output. Other expedients for reducing the weight of engines are the use of sheet metal jackets, block cylinders, or cylinders so cast as to permit of their being bolted together to form a block; hollow crankshafts, sheet metal crank bases, cast steel flywheels of large diameter and tubular or I-beam construction for all parts subjected to compression and bending stresses.

CHAPTER XX

Sleeve-Valve Engines

Up to about 1908 the only form of valve that had proven commercially successful on gasoline engines was the poppet valve. Dr. Otto, the inventor of the four-cycle gas engine, in one of his earliest machines used a sliding valve, and all the different forms of valves used in steam engines, including flat slide, piston and rotary valves, were at one time or another applied to gasoline motors, but they never proved successful in the long run. In 1908 Charles Y. Knight, of Chicago, induced the Daimler Motor Co., of Coventry, England, to take up an engine of his invention in which the valve functions are performed by two ported sleeves one within the other, and both between the piston and the cylinder wall. The two sleeves are reciprocated by a half-speed shaft carrying small cranks equal in number to the sleeves. This engine has since proven very successful, and has been taken up by leading manufacturers in different European countries and in the United States.

The Knight Engine—Figure 340 is a sectional view through one of the cylinders of a Knight engine. Within the cylinder bore there is a thin sleeve A of cast iron, cut with ports on opposite sides for the inlet and exhaust. Within this sleeve there is another, B, also cut with ports near its upper end, and within the latter sleeve is located the piston. In these engines the cylinder head C is detachable, and is of rather unusual construction. A part of it extends into the cylinder for quite a distance, but is of a diameter smaller than the cylinder bore, so the two sleeves may enter between it and the cylinder wall. Both the cylinder head and the piston head are concave or bowl shaped, the object being to secure as nearly as possible a spherical combustion chamber. The two sleeves are reciprocated by means of a half-speed shaft, which is driven from the crankshaft through a silent chain. The half-speed shaft is formed with eight cranks, and these cranks connect through short connecting rods EE with lugs formed on the valve sleeves at their lower ends. The connecting rods for the two sets of sleeves are of different lengths, and usually they are slightly inclined, that is to say, the vertical line along which their upper end reciprocates does not pass through the axis of the half-speed shaft, but is somewhat closer to the cylinder axis—this for the sake of compactness. It will be noticed that the drive of the sleeves is unsymmetrical, a feature the practicability of which was questioned at one time, but it has been found that owing to the long bearing of the sleeves it does not cause any trouble. At the lower ends, where the driving lugs are, the sleeves are suitably strengthend by circumferential flanges.

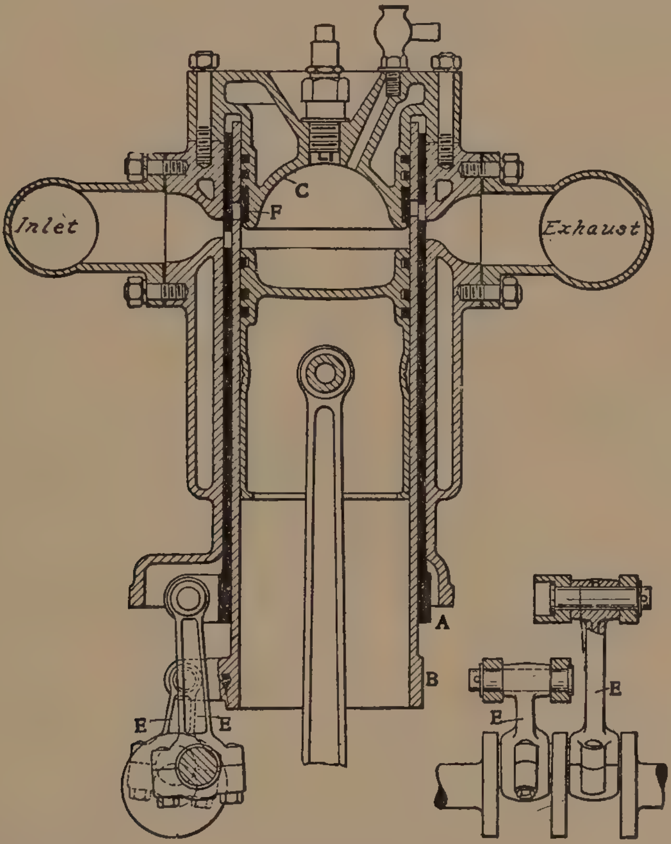


Fig. 340.—Section of Knight Double-Sleeve-Valve Engine.

One peculiarity of the Knight engine is that the bore of the cylinder casting is not the effective bore of the engine, the latter being equal to the bore of the inner valve sleeve. The sleeves are made of a thickness equal to about 0.040 in. per in. of effective bore. The outer sleeve is sometimes made slightly thinner than the inner one, because it is not subjected to the pressure of the explosion. Thus, for instance, in the 100 × 140 mm. Panhard-Knight engine the inner sleeve was 5 mm. and the outer sleeve 4 mm. thick. The bore of the cylinder casting in this engine therefore was 118 mm.

Packing Ring Arrangement—Gas tightness is insured by two sets of packing rings, one set of three or four being placed on the piston in the usual way, and the other set on that portion of the cylinder head which extends into the cylinder and is surrounded by the valve sleeves. The latter set comprises one unusually wide ring, F, at the bottom, referred to as a junk ring, and two or three rings of the same dimensions as used on the piston. The object of the junk ring is to seal the ports in the valve sleeves while the explosion takes place and the gases expand in the cylinder.

In most designs of Knight engine the ports in the inner sleeve are of great circumferential width, and to prevent the junk ring from catching on the edge thereof, it is made wider than the ports. In Fig. 340 the moving parts are shown in the position corresponding to the beginning of the power stroke, and it will be seen that both the inlet and exhaust ports in the inner sleeve are fully covered by this junk ring.

Port Dimensions—Each port extends substantially one-third around the cylinder. The ports are provided with a bridge at the middle of their length so as to prevent undue weakening of the sleeve. On the theory that the port area should be proportional to the piston head area, the height of the ports should be proportional to the effective bore or piston diameter. Most of the earlier Knight engines were of about 4 in. bore and slightly over 5 in. stroke, and in these the inlet port was ½ in. and the exhaust port ⅝ in. high.

Valve Motion and Timing—The valve action and the effect of various factors upon the valve timing can best be studied by means of the diagrams, Figs. 341 and 342. It may be stated that the small cranks for driving the two valve sleeves are in practice set so as to make an angle of 60° to 90° with each other. In the diagrams it is assumed that they are set at an angle of 80°. It is further assumed that the half-speed shaft is directly underneath the hinge joint to the lug of the sleeve, and that the connecting rods are of twice the length of the valve motion. We have here a regular crank and connecting rod mechanism, the same in every particular as the main crank and connecting rod of the engine, and the laws of motion previously deduced for this mechanism (equations 24 and 56) therefore apply in this case.

In drawing Figs. 341 and 342 the sleeve travel was assumed to be 1⅛ in. It will be noticed that in each diagram there are four curves, representing the up-and-down motion of the top and bottom edges of the inlet and exhaust ports in the sleeves respectively. In Fig. 341 there are also two horizontal lines representing the top and bottom edges, respectively, of the inlet port in the cylinder wall. The inlet begins to open when the bottom edge of the port in the outer sleeve, moving downwardly, passes the top edge of the port in the inner sleeve moving downwardly. The inlet closes when the bottom edge of the port in the inner sleeve, moving upwardly, passes the top edge of the port in the outer sleeve also moving upwardly. By reference to the scale at the bottom of the diagram it will be seen that the inlet extends over 200° of crank motion. Now, it is customary to open the inlet 10° to 15° after the crankshaft passes the upper dead center and to close it about 30° to 35° after the crankshaft passes the lower dead center. Since the total opening extends over 200°, or 20° more than a half revolution, it is evident that the lag of inlet closing must exceed the lag of inlet opening by 20°. We will therefore let the inlet open 10° late and close 30° late. The point on the horizontal axis at which the inlet begins to open therefore corresponds to a crankshaft motion of 10° from the top dead center. This enables us to plot the four dead center positions on the horizontal scale, and to properly divide the whole scale.

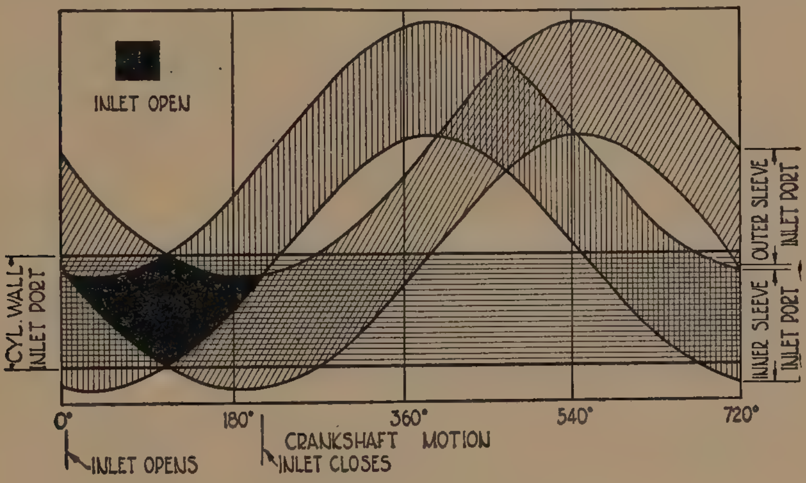


Fig. 341.—Diagram of Inlet Action.

Figure 342 is a diagram of the exhaust port action. The exhaust begins to open when the lower edge of the port in the inner sleeve, moving downwardly, passes the lower edge of that portion of the cylinder head which extends into the cylinder. The exhaust closes when the top edge of the port in the outer sleeve, moving downwardly, passes the lower edge of the port in the cylinder wall. In order to secure the proper length of valve opening the ports in the inner and outer sleeves are placed at different levels, the one in the outer sleeve being at a lower level (¾ in. lower in the diagram). The exhaust opening under the assumptions here made extends over a period corresponding to 240° of crank motion. The exhaust begins to open 50° ahead of the lower dead center and closes 10° past the top dead center. In order to effect this valve timing the half-speed shaft must be so set that when the engine crankshaft is in the top dead center position at the beginning of the inlet stroke, the crank operating the inner sleeve has to travel through 15° until it reaches the bottom dead center position, and the crank operating the outer sleeve has to travel through 95° till it reaches the bottom dead center position.

Diagrams like the above are very convenient for studying the effect of various changes in the ports and valve gearing, as, for instance, a change in the angle between the cranks for the two sleeves of each cylinder, a change in the width of ports, etc. It may be pointed out that the various licensees under the Knight patent design their engines in their own way. In Mercedes-Knight engines the angle between the two sets of sleeve-operating cranks is 70°.

Port Opening Areas—While the chief object originally aimed at by the inventor was silent operation, it was also claimed for the Knight engine that it enabled unusually large valve capacity to be provided and high volumetric efficiency and high mean effective pressures to be obtained in consequence. It may be of interest to compare the valve capacity of a Knight engine with that of a poppet valve engine. Suppose an engine of 4 in. bore to have inlet ports ½ in. high and extending one-third around the circumference. Also, let the valve timing be that depicted in Fig. 340. Then the opening of the valve is represented by the diagram Fig. 343. This diagram is practically a triangle, the opening curve being very slightly concave and the closing curve convex (from the inside). The average height of opening is therefore ¼ in. The length of the port around the circumference is

$$\frac{4 \times 3.14}{3} = 4.19 \text{ in.}$$

and the valve capacity therefore is

$$4.19 \times \tfrac{1}{4} \times 200 = 209.5 \text{ sq. in. deg.}$$

Now let us see what diameter of poppet valve would be required to equal this. Assume that the lift of the poppet valve is one-fifth of the clear diameter. Also assume that the valve is opened with constant acceleration and deceleration in 85° of crankshaft motion, and that the total opening period is 200° of crank motion. Then the equivalent duration of full opening is

$$200 - 85 = 115°,$$

and the full opening area of the valve must be

$$\frac{209.5}{115} = 1.82 \text{ sq. in.}$$

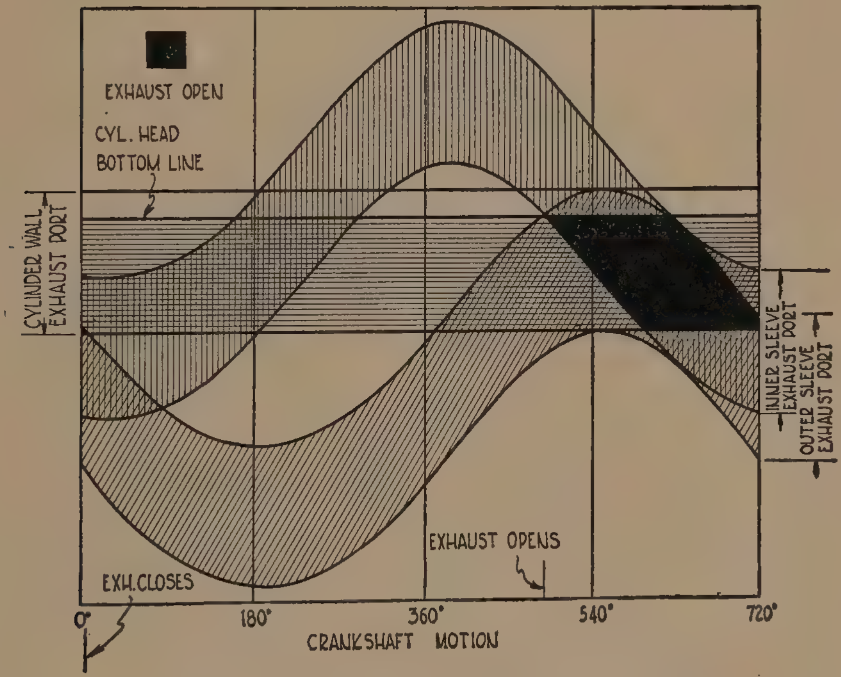


Fig. 342.—Diagram of Exhaust Action.

Now, according to equation (67) the area of opening of a poppet valve is

$$A = \pi(0.7dh + 0.35h^2)$$

and with

$$h = \frac{d}{5},$$

$$A = \pi(0.14d^2 + 0.014d^2) = 0.484d^2.$$

We have, therefore,

$$0.484d^2 = 1.82 \text{ sq. in.}$$

$$d = \sqrt{\frac{1.82}{0.484}} = 1.94 \text{ in.}$$

A diagram showing the variation of the opening area of a poppet valve of 1.94 in. clear diameter is superposed on the sleeve valve diagram in Fig. 343. The two diagrams should be of the same area. It appears that the sleeve valve diagram has a slightly greater area, which is probably due to a slightly erroneous assumption in making the calculation.

It is frequently said that the sleeve valve opens much quicker than the poppet valve, but this is only partly correct. The fact is that the sleeve valve opens and closes at substantially uniform speed, whereas the poppet valve begins and completes the opening motion at practically no speed and attains its maximum opening speed at half opening. The sleeve valve opens the most rapidly at first but the least rapidly later on, and the poppet valve attains its full opening in a shorter time than the sleeve valve.

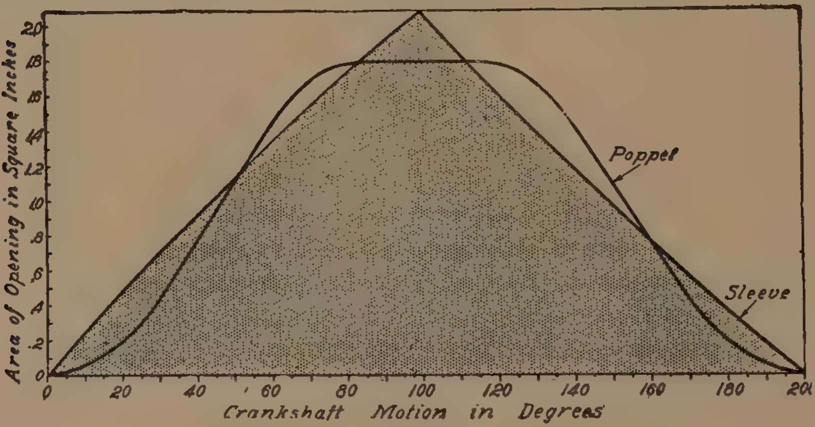


FIG. 343.—DIAGRAM OF INLET-OPENING AREA.

Figure 344 shows the opening diagram of the exhaust ports for a sleeve valve, the maximum width of the port opening being ½ in., the length 4.19 in. and the timing as in Fig. 342. It will be seen that the port attains its maximum opening in 103° of crankshaft motion, remains fully open for 27° and closes 240° of crankshaft motion after beginning to open. The valve capacity, therefore, is

$$\frac{27 + 240}{4} \times 4.19 = 280 \text{ square inch-degrees.}$$

In order that a poppet valve having a lift equal to one-fifth the clear diameter, which is accelerated and decelerated uniformly and attains its full lift in 85° of crank motion, may have an equal valve capacity, it must have a diameter d such that

$$0.484d^2 = \frac{280}{240 - 85} = 1.807 \text{ sq. in.}$$

$$d = \sqrt{\frac{1.807}{0.484}} = 1.93 \text{ in.}$$

It may be said that comparatively few poppet valve stock engines are fitted with such large valves.

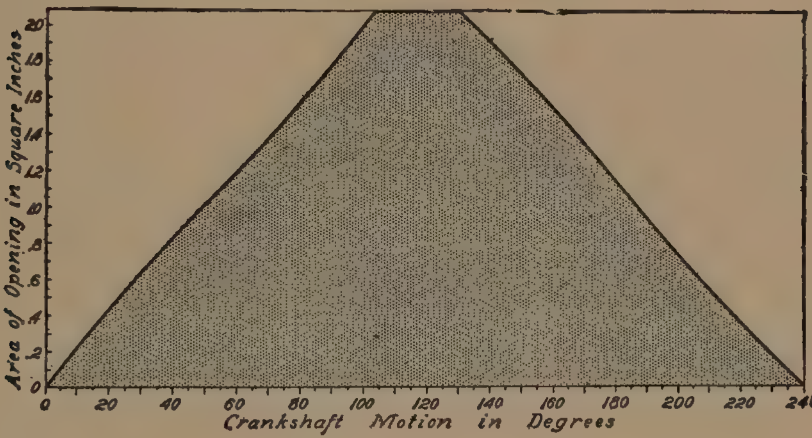


FIG. 344.—DIAGRAM OF EXHAUST-OPENING AREA.

Characteristics of Knight Engine—In respect to silence of operation, which was its outstanding feature when first developed for the market, the Knight engine is now equalled by poppet valve engines of the better grades at least. More importance is now attached by its manufacturers to the fact that its output increases in the course of use, for some time at least. This, no doubt, is due to the fact that the sleeves and packing rings wear down to a better fit, whereby the frictional losses are reduced and the seal of the combustion chamber is improved. Tests of the power consumed for operating the sleeves and accessories (pump and magneto) were made by Prof. Riedler, who found the power loss due to this cause to be 2 hp. when the engine developed 40 hp. on the brake.

Lubrication—The early Knight engines had splash lubrication, like most other engines of the period. Dippers on the connecting rod head threw oil from splash troughs onto the lower ends of the sleeves, and the suction in the cylinder during the inlet stroke drew it upward to the upper ends of the sleeves. To aid in the proper distribution of the oil the sleeves had numerous oil grooves cut on their outside, and oil holes were drilled through their walls, generally along helical lines, so that the oil could pass from one side of the sleeve to the other. It was found that some regulation of the splash was necessary or too much oil would be drawn up into the combustion chamber when the engine was idling or lightly loaded, and the engine would give off a smoky exhaust. The splash troughs were therefore pivoted on one side and had their other sides connected to the throttle mechanism by a suitable linkage, so that as the throttle was opened and the load on the engine increased, the troughs were raised and more oil was splashed around.

Recent Improvements in Knight Engines—Up to 1923 all Knight type engines had cast iron sleeves of a minimum thickness of about ⅛ in. As the speeds of passenger-car engines in Europe were getting higher and higher, something had to be done to reduce the weight of the sleeves, and steel sleeves were adopted about that time by both Panhard & Lavassor of France and the Daimler Motor Co. of England. Panhard now uses sleeves only 1.5 mm. (0.060 in.) thick, the outer sleeve being given a thin lining of white metal to provide a good bearing for the inner sleeve. With steel sleeves the Daimler engines, of 5-in. stroke, run at 4000 r.p.m. The cast iron and steel sleeves of the Panhard Knight engine are shown side by side in Fig. 345.

The considerable thickness of the cast iron sleeves was objectionable both because of the resulting inertia loads on the eccentric shaft bearings and the unbalance due to the sleeves at high speeds. In a four-cylinder engine, for instance, while the primary forces on the pistons and associated reciprocating parts are balanced, those on the sleeves are not balanced, because these reciprocate at only one half the rate. The pistons in the two end cylinders, for instance, are in phase with each other, but the sleeves of the same kind in the end cylinders are 180° out of phase. This produces a rocking couple in the fore-and-aft plane. On the other hand, the secondary forces on the sleeves of a four-cylinder engine are balanced or neutralized, because these have the same period as the primary forces on the pistons. But the primary forces are several times as large as the secondary forces.

Panhard eliminated the wide junk ring on the cylinder head by providing the ports in the inner sleeve with a number of narrow bridges, which permits of using on the cylinder head, rings of the same width as on the pistons. It was the necessity of having the port width in the vertical direction less than the width of the junk ring that limited the size of the ports, and with the introduction of steel sleeves the port widths have been greatly increased, permitting of the attainment of much higher speeds. The eccentric throw, of course, must be increased in the same proportion as the width of the ports.

An incidental advantage of the substitution of narrow for the wide junk rings was that the distance which the sleeves extend above the bottom edge of the cylinder head could be reduced and the height of the engine decreased. The length of the sleeves was further decreased by cutting a large slot in the outer sleeve at the bottom, in which moves the lug on the inner sleeve. It is therefore no longer necessary that the inner sleeve should project below the outer one a distance equal to the width of this lug. The inner sleeve, moreover, is also provided with a slot at its lower end, into which the connecting rod swings at its moment of greatest angularity.

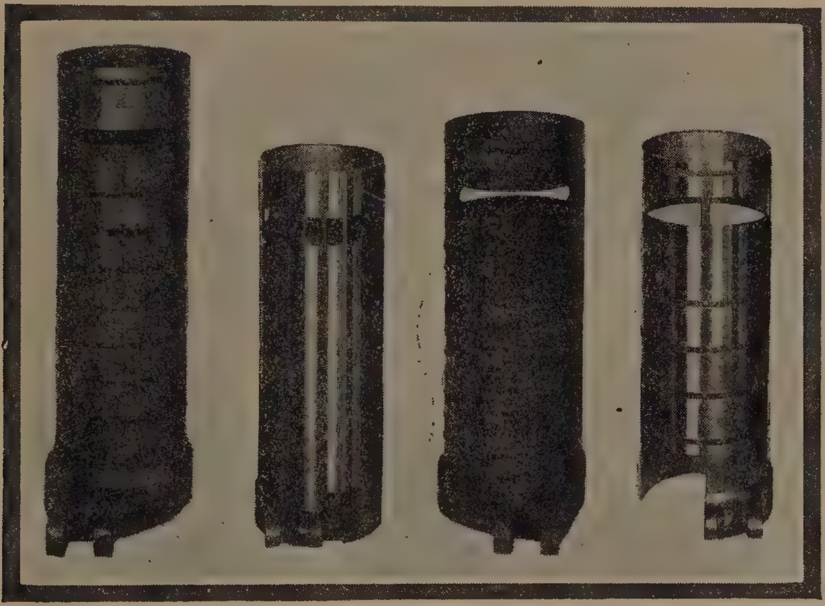


Fig. 345.—Comparison of Cast Iron and Steel Sleeves.

The increase in the speed of these engines made necessary the use of pressure lubrication, so that the method formerly employed for regulating the supply of lubricant to the cylinder walls, by means of swiveled splash troughs, was no

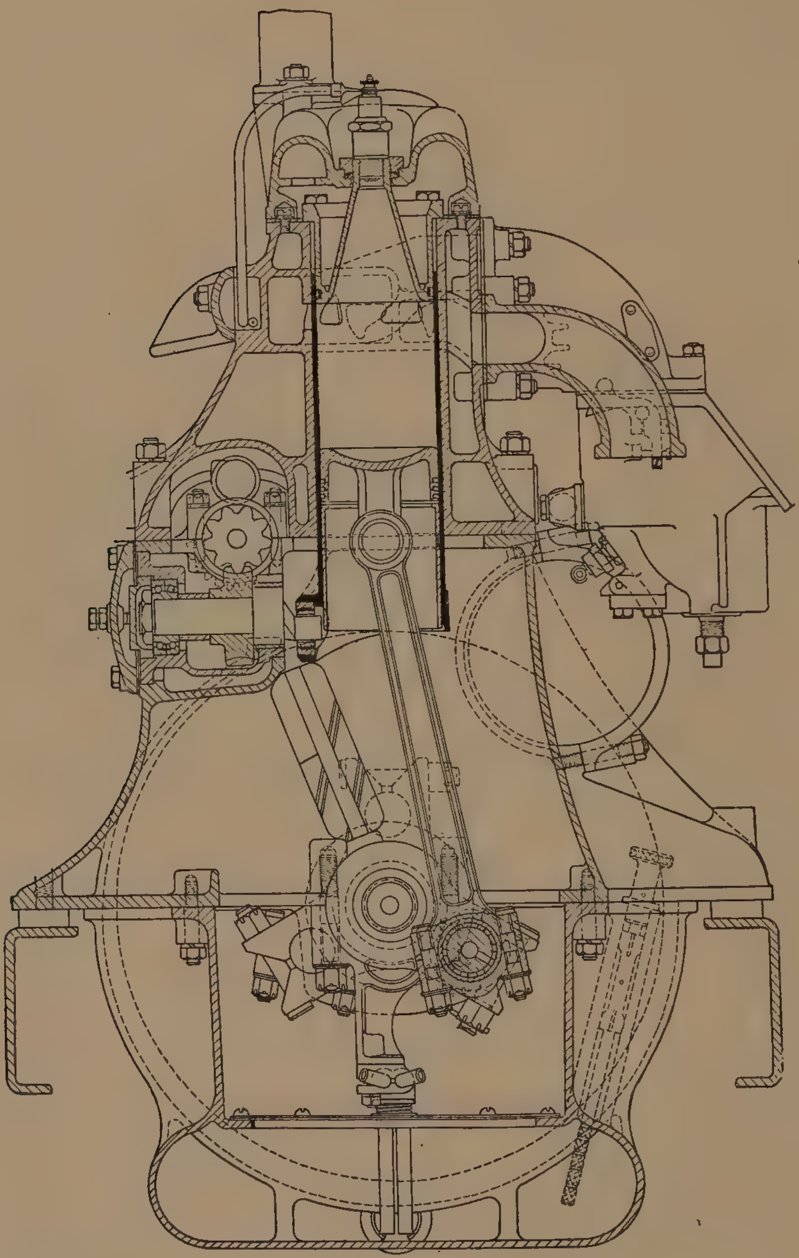


FIG. 346.—CROSS SECTION THROUGH BURT SINGLE SLEEVE VALVE (VAUXHALL DESIGN).

longer applicable. A similar effect is now obtained by providing an extra oil lead to the upper part of the cylinder wall, the delivery of which is controlled by a connection to the throttle valve. Oil is delivered into the upper part of the engine through this lead only when the throttle valve is nearly fully open.

Single Sleeve Valve Engine—A single sleeve valve engine was invented by a Scotch engineer, P. Burt, and has been in service almost as long as the Knight engine, the first applications having been made in 1911. It was used originally by the Argyll Company of Glasgow, but is now used also by the Vauxhall Company of England and several Continental firms. The rights to this engine are controlled by the Continental Motors Corp. of Detroit, and passenger-car engines of this type were placed on the American market early in 1926.

In the Burt engine a single sleeve is inserted between the cylinder wall and piston, which has what amounts to a combination reciprocating and oscillating motion. Referring to Fig. 346, from the crankshaft is driven a timing shaft in a 1:1 ratio. This timing shaft, running lengthwise of the engine, carries as many pinions with helical teeth as there are cylinders. Each pinion meshes with a helical gear of twice its number of teeth, with its axis across the engine and intersecting the cylinder axis.

The helical gear is mounted on a shaft carrying a small crank, which latter engages with a lug on the valve sleeve, through the intermediary of a sliding sleeve with spherical seat in the lug. Every point on the crankpin, of course, has a circular motion, but the bearing slides in and out on the pin and therefore traces a curve which forms the intersection of one cylindrical surface with another one of larger diameter. However, owing to the fact that the radius of the sleeve is less than the distance from the axis of the sleeve to the center of the spherical bearing, the horizontal motion of the sleeve is less than that of the bearing. When viewed from the side, any point in the sleeve has an elliptical motion, with the long axis of the ellipse vertical. Inlet and exhaust ports are formed in the cylinder wall, on opposite sides of the engine, and there are corresponding ports in the sleeve, which come into registry with these cylinder ports at the proper moments. The exhaust port in the cylinder wall is indicated in Fig. 347 by the shaded area, while the corresponding sleeve port is shown unshaded, in three positions, at the beginning, middle and end of the exhaust period, respectively. The port in the sleeve follows the path indicated by the ellipse. In the upper position of the sleeve port it is about to come into registry with the cylinder port and the exhaust period begins (50° ahead of bottom dead center). In the next position the exhaust is fully open, the doubly cross-sectioned area being the effective port opening area). In the lowest position of the sleeve port registry with the cylinder port ceases and the exhaust period ends. While the sleeve port has traveled from the upper to the lower position the crank which moves the sleeve has described an arc of 120°, and the crankshaft, of course, during the same period has described an arc of 240°; hence, since the port began to open 50° before the bottom dead center, it will close 10° after top dead center. A point worth noting is that at the moment the exhaust period begins the sleeve port moves nearly straight down, and as it moves across the upper edge of the cylinder port with its full width at the same time, the opening of the exhaust is quite rapid, which permits of a rapid evacuation of the cylinder.

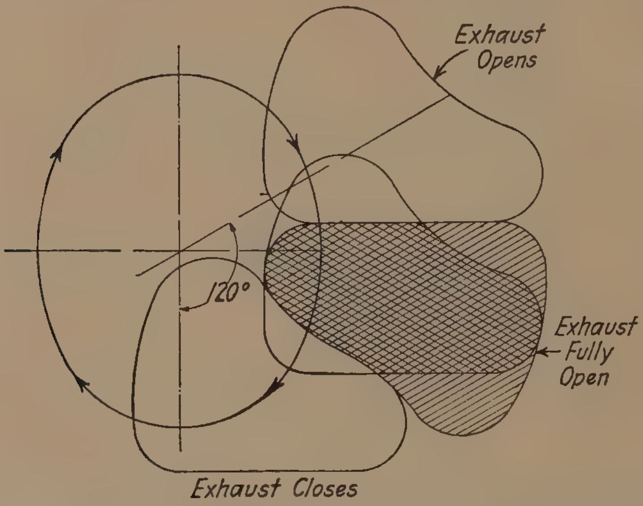


Fig. 347.—Exhaust Action of Single Sleeve Valve.

The form and action of the inlet ports are quite similar, as may be seen from Fig. 348. The inlet port in the cylinder wall (cross hatched) is reversed as compared with the exhaust port, its flat side being the closing side. In the lowest position of the sleeve port it is about to come into registry with the cylinder port and the inlet period is therefore about to begin. In this particular layout, this corresponds with the closing of the exhaust port (10° after top dead center). In the next position of the sleeve port in the diagram the inlet port is fully open, the effective opening area being indicated by double cross-hatching, and in the following position the inlet port closes. The inlet period is equal to 110° of sleeve crank motion, instead of 120°, due to the fact that the vertical height of the inlet ports is less than that of the exhaust ports. Therefore, since the inlet begins to open at 10° past top dead center, it remains open until 50° past bottom dead center.

The whole of the space around the cylinder is utilized for ports, and there are usually three inlet and two exhaust ports in the cylinder wall, and four ports in the sleeve. One of the ports in the sleeve is of special form, as shown in Fig. 349, and registers with both an inlet and exhaust port in the cylinder wall. The spacing between ports of the same kind in the cylinder wall is such that the sleeve port can pass between them without registering with either of them, hence the width of the individual port can be made about one-ninth the circumference of the cylinder bore, and the height of the exhaust port will be about two-thirds the throw of the sleeve-operating crank. The valve opening area diagrams are substantially triangular in shape for both inlet and exhaust.

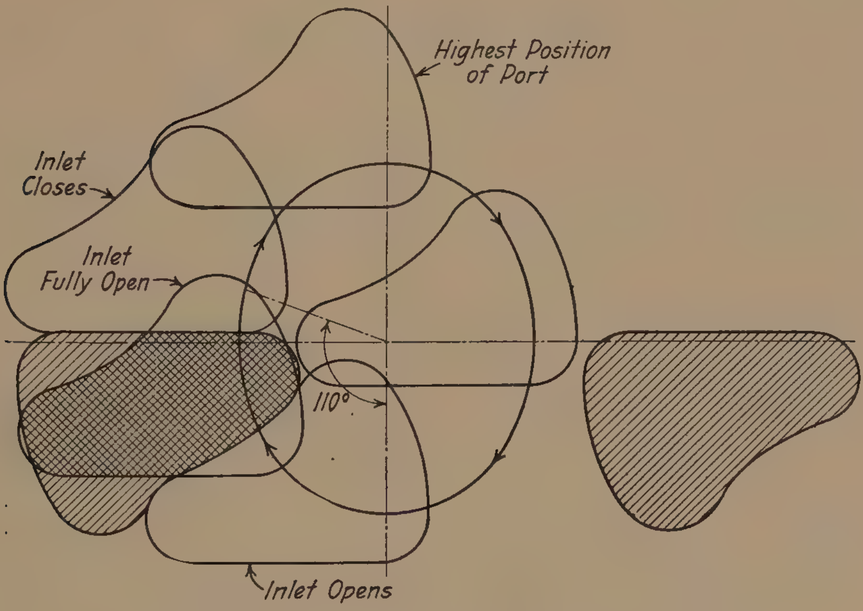


Fig. 348.—Inlet Action of Single Sleeve Valve.

A compression ratio as high as 5 has been used with Burt type single sleeve valve engines, the favorable form of the combustion chamber making this practical, it is claimed. With the design shown in Fig. 346 the piston comes practically up to the cylinder head at the end of the compression stroke, thus creating considerable turbulence in the compression chamber at this moment.

Piston and Rotary Valve Engines—The success of Knight with his sleeve-valve engine induced many other inventors to devote their efforts to the development of positively-operated valves for gasoline engines. Most closely related to the sleeve valve is the so-called cuff valve, consisting of a short split sleeve fitting inside the cylinder bore above the upper end of the piston and reciprocated by means of a stem extending through the cylinder head. Then, engines have been built with two semi-cylindrical slides between piston and cylinder wall, reciprocated independent of each other and serving for the inlet and exhaust respectively. In another design the slides were of crescent-shaped cross-section and required guide surfaces to be bored and finished before the cylinder bore proper was finished.

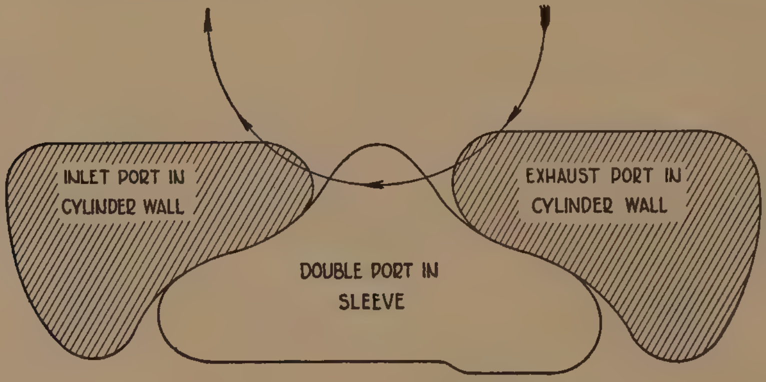


FIG. 349.—DOUBLE PORT IN SLEEVE.

Piston-valve engines were suggested in which there were two cylindrical valve chambers parallel with the working cylinder, on opposite sides, of a bore almost as large as the working cylinder, containing reciprocating trunk pistons with ports in their skirts, adapted to register with ports in the cylinder wall during the proper periods.

Still more work was done on rotary valves of both the plug and disc type. Some of these engines were used on cars in regular production for several years but not one of them has endured.

CHAPTER XXI

The Two-Stroke Cycle Engine

The two-cycle engine has been used very extensively for marine work, but its use on automobiles has always been limited. In fact, at the present time (1926) there is not a single concern in the United States manufacturing cars with engines of this type though it is used on two makes of motorcycles.

The characteristic feature of the two-cycle engine is that there is an explosion in each cylinder during every down stroke of the piston, instead of during every second down stroke, as in the four-cycle engine. One other feature which distinguishes the two-cycle engine from the four-cycle is that the combustible charge must be pre-compressed in order to get it into the working cylinder. This is generally done in the crankcase, which is made gas-tight and acts as a pump; but it is sometimes done by means of a pump formed by extensions of the working cylinder and piston or by an entirely separate pump cylinder. As originally adapted from marine practice the two-cycle engine was of extreme simplicity, but it possessed certain defects, and in order to overcome these it was found necessary to add parts, thus complicating it to a certain extent.

The Two-Stroke Cycle—The successive operations in the cylinder and crankcase of a two-cycle engine are as follows: Referring to Fig. 350, while the piston is moving up in the cylinder it creates a partial vacuum in the crank chamber, and when it reaches a certain point in its up-stroke it uncovers a port A in the wall of the cylinder, through which combustible charge from the carburetor is drawn into the crank chamber. During the following down-stroke, after the port A has been covered by the piston, this combustible charge is slightly compressed in the crank chamber, the usual compression pressure being between 6 and 8 lbs. per sq. in. gauge. As the piston approaches the bottom end of its stroke it uncovers a transfer port B in the cylinder wall (see Fig. 351), and the combustible charge under pressure in the crank chamber flows through the transfer passage into the cylinder. Shortly after the piston starts on its up-stroke it closes the transfer port, and during the remainder of the up-stroke the charge is compressed in the cylinder. When the piston reaches the top end of the stroke the charge in the cylinder is ignited and the piston is forced down. As the piston approaches the lower end of its stroke it uncovers the exhaust port C in the cylinder wall opposite the transfer port B (see Fig. 352), and the burnt gases, which are still under considerable pressure, escape through this port. Since the exhaust port opens slightly earlier than the transfer port, a considerable portion of the burnt gases escapes before any new charge enters. The rest of the burnt gases are swept out by the incoming fresh charge. It is, of course, impossible to insure that all of the burnt gases be swept out and that none of the

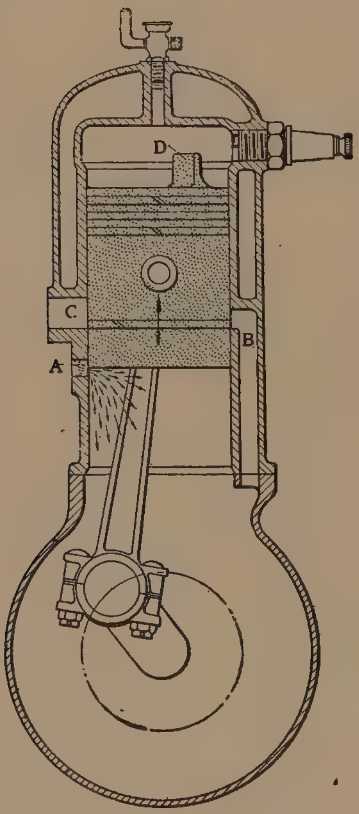


Fig. 350.—Admission to Crankcase.

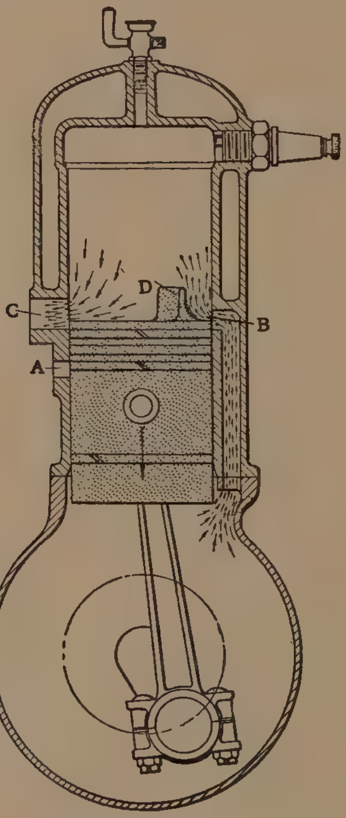


Fig. 351.—Transfer of Charge.

fresh charge escape; but in order to prevent as far as possible the mingling of the fresh charge with the spent gases a deflector D is formed on the piston head opposite the transfer port which deflects the incoming current of fresh gas upwardly (Fig. 351).

The evacuation of the cylinder and the transfer of a new charge from the crank chamber to the cylinder therefore go on simultaneously; and, besides, the periods of admission and exhaust, measured in degrees of crank motion, are much shorter than in the four-cycle engine. While in a four-cycle engine the admission extends over a period corresponding to about 200° of crank motion, and the exhaust to a period corresponding to 230° of crank motion, in a two-cycle the inlet extends over less than 90° of crank motion, and the exhaust over less than 120°.

Crankcase Inlets—In the older designs of two-cycle engine the charge from the carburetor was admitted to the crank chamber through a suction operated poppet valve. This type is known as the two-port engine, and has been extensively used for marine work. It never found any application for automobile purposes, owing to the fact that it does not lend itself to operation at high speed. Since the degree of vacuum in the crank chamber is always quite low, the valve actuating force is very small, and at high speeds the inertia of a suction actuated valve would be an impediment.

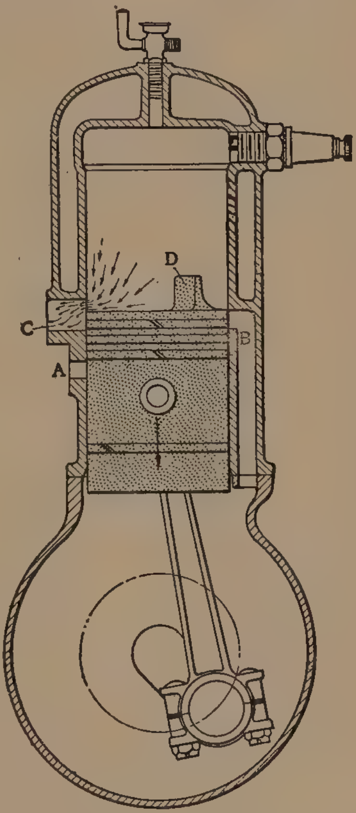


Fig. 352.—Exhaust.

The type of two-cycle engine illustrated in Figs. 350-352 is known as the three-port type. In it a port is cast in the cylinder wall at such a point that it will be uncovered by the lower end of the piston when the latter approaches the top end of its stroke, and since at that time there is a partial vacuum in the crank chamber, combustible charge from the carburetor is drawn through the port into the crank chamber. Such a port opens and closes positively, but it remains open only a comparatively short length of time. The suction on the carburetor starts and stops abruptly, which condition calls for a special design of carburetor.

Positive operation can be combined with gradual opening and closing by using a disc type of valve on the crankshaft, a port in the disc registering with a port in the crankcase wall during part of the revolution. A variation of this plan consists in providing one of the crank journals (which is made hollow) with a port registering with a similar port in the bearing.

Crank Chamber Design—The crank chamber of a two-cycle engine must be made as small as possible, or the necessary crank chamber compression will not be obtained. If we assume that the crank chamber fills to a pressure of 14 lbs. per sq. in. abs. and we wish a precompression of 7 lbs. per sq. in. gauge or 21.7 lbs. abs., then the required crank chamber volume figures out to 80 cu. in. In order to get the volume down as small as this various expedients are resorted to. In the first place, the crank chamber wall is made to come as close as possible to the form described by the crank in its revolution. Then the connecting rod is made as short as possible, say, 1.8 times the stroke, and if this does not bring the volume down sufficiently a crankshaft with discs instead of crank arms may be used.

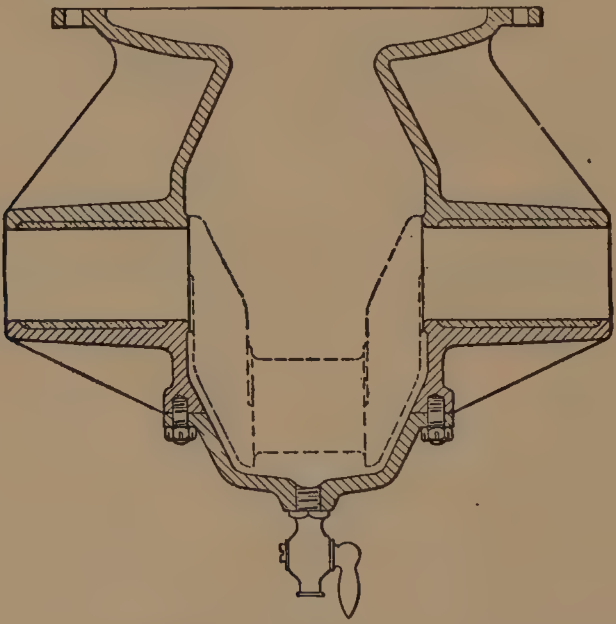


Fig. 353.—Crankcase Design for Two-Stroke Engine.

A manograph diagram from the crankcase of a two-cycle engine is shown in Fig. 354. At the beginning of the upstroke, there is a slight over-pressure in the crankcase, and this increases during the first part of the stroke, as long as the transfer port remains open, which may be due to the heating of the charge as it comes in contact with the cylinder walls, or to a return wave following the surging of the charge into the cylinder. Shortly after the transfer port is closed a vacuum begins to form in the crank chamber, increasing up to the point where the inlet port opens. The crank chamber then fills rapidly and at the end of the stroke is filled nearly to atmospheric pressure. During the return stroke the charge is compressed in the crank chamber and attains a maximum pressure of about 8.5 lbs. per sq. in. at the point where the transfer port begins to open. The pressure thereafter rapidly drops to near atmospheric.

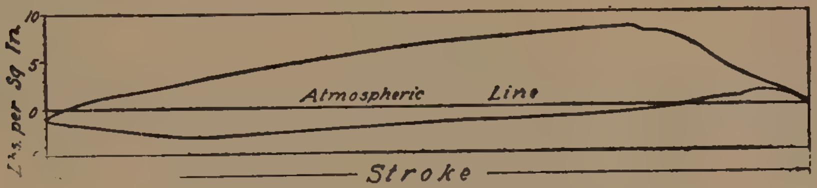


FIG. 354.—CRANKCASE-PRESSURE DIAGRAM.

Since the crank chamber acts as a pump it must be made and maintained air-tight. In the three-port type of engine it is necessary to place one packing ring on the piston near the lower end, so that the charge under pressure in the crank chamber will not flow back by the piston and out through the inlet port. There is also a chance for leakage through the crankshaft bearings. In order to minimize this leakage the bearings must be made of liberal size, and particularly of good length, so they will wear very slowly. Grease lubrication of the bearings tends to keep the charge in, but is unsuitable for automobile work. Force feed of oil to the bearings is the next best thing. Some manufacturers provide the end bearings with packing on the outer ends.

In a multicylinder two-cycle engine the crank chamber of each cylinder must be separate, in order that it may act as a charge pump.

Transfer Passage and Deflector—From the crank chamber the charge must be transferred to the cylinder while the piston is at and near the lower end of its stroke. Sometimes the transfer passage extends down the side of the cylinder and through the top portion of the crank chamber, as seen in Figs. 350-352. However, certain designers prefer to shorten the passage by terminating it below in a port in the cylinder wall with which registers a port in the piston wall when the piston is at the bottom end of its stroke. (See Fig. 355.)

The cross-section of the transfer passage should be at least equal to that of the transfer port. All of the three ports extend over 80° to 85° of the cylinder circumference, but their effective width is reduced by a bridge at the middle about 3/8 in. wide. This bridge is necessary in order to prevent the piston rings from springing out and being caught by the edges of the port. The bridge in the exhaust port naturally becomes very hot and in cylinders of considerable size it is customary to make it hollow and cool it by either water or air circulation.

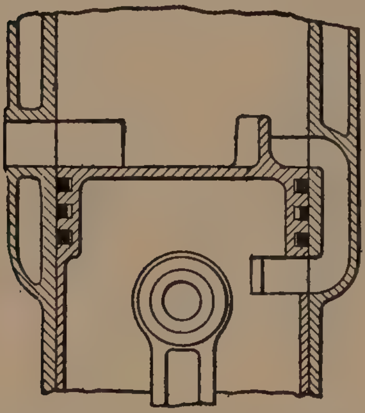


Fig. 355.—Transfer Port in Piston Wall.

It is a good plan to insert a wire screen in the transfer passage or place it against the opening of same into the crank chamber, so as to prevent explosions in the crank chamber.

In order to prevent the fresh charge from mingling with the spent gases, a deflector plate is usually formed on the piston head. The form of this deflector differs considerably in different engines. Its height is generally limited by the height of the compression space. Sometimes, in order to increase the permissible height, the deflector is made part of the piston head; that is, the piston head slopes down from the upper end of the deflector.

Port Sizes—The operation of a two-cycle engine depends very largely upon the dimensions and location of the different ports. The ports must be dimensioned and located very accurately and they are usually finished by milling. In a crankcase-compression type of engine the transfer-port opening is made to extend over about 90° of crank motion and the exhaust port opening over 115°.

Differential Pistons—To avoid the difficulties inherent in crankcase precompression, differential pistons have been used to some extent. This construction is applicable only in the case of engines with an even number of cylinders. The cylinder is made with a small bore at the top end and a larger bore at the lower end. The piston is also made with a double diameter, the small diameter portion on top fitting the small bore of the cylinder and the larger diameter lower portion fitting the bigger bore. The regular working cycle takes place in the upper end of the cylinder above the piston head. An annular space is formed between that portion of the cylinder wall which is bored to the larger diameter, the small diameter portion of the piston and the shoulders or offsets in the piston and cylinder walls, respectively. As the piston moves down the combustible charge is drawn into this annular space, and on the return stroke the valve or port closes and the charge is compressed in the annular space to a relatively high pressure. The ports would be so proportioned that at the moment the transfer port is opened the distance of the offset in the piston from the offset in the cylinder is equal to about one-quarter the length of stroke. When the piston approaches the end of its up-stroke the annular space is placed in communication with the upper end of another cylinder in which the piston is at that moment approaching the end of its down-stroke, and which is, therefore, ready to receive a new charge. This method of construction is particularly applicable to double cylinder engines. In these the two cranks are set at 180°, and one piston is at the top end of the stroke while the other is at the bottom end. The port in the wall of the annular chamber of one cylinder communicates through a passage with the working end of the other cylinder, and when the transfer port of the latter cylinder is uncovered by its piston the charge from the annular compression space is transferred to the working end of the other cylinder. The displacement in the annular space would generally be made a little greater than the displacement in the working end of the cylinder, with the object of supercharging the latter.

A very complete series of tests of an ordinary two-stroke engine was made by Dr. W. Watson, of London, and reported by him in a paper read before the Institution of Automobile Engineers. The engine was a single-cylinder one with a bore and stroke of 3¼ in. each. It employed crankcase compression and was of the three-port type. The crankcase inlet port remained open during 82° of crank travel, the transfer port during 97°, and the exhaust port during 122°. This engine at 1506 r.p.m. developed 4.16 b.-hp., with a fuel consumption of 0.76 lb. of gasoline per indicated hp.-hr. The mean effective pressure at 1500 r.p.m. was 53 lbs. per sq. in. At 900 r.p.m. it was 65 lbs. per sq. in. The ratio of air to gasoline at this speed was 12.13 to 1, and 17 per cent of the charge supplied to the cylinder escaped through the exhaust valve. The loss of charge was much greater at low speeds. The compression in this engine was between 62 and 65 lbs. per sq. in. at all speeds.

Although some thirty models of motorcycle manufactured in different parts of the world are equipped with two-stroke engines, only three of those built in the U. S. in 1925 had engines of this class. The engine of one of these is shown in section in Fig. 356. It has a bore of 2¾ and a stroke of 2⅞ in. and is rated at 2¾ hp.

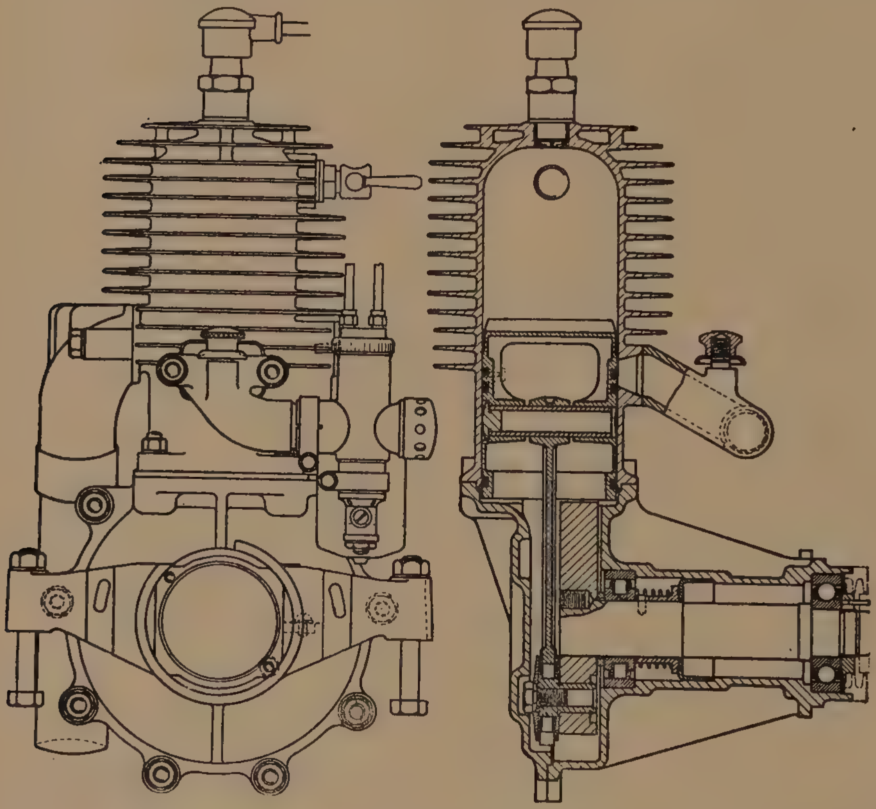


Fig. 356.—Elevation and Cross-section of Neracar Two-Stroke Engine.

CHAPTER XXII

Diesel-Type Automotive Engines

Since about 1920 much work has been done on small-size Diesel engines with a view to adapting them for automotive uses. A number of firms, particularly in Germany, are now manufacturing them for trucks, tractors and rail cars, and there is little doubt that the use of these engines for automotive purposes will increase, especially in countries where volatile liquid fuels are relatively expensive.

Diesel Engine Fuels—The chief advantage of the Diesel engine is that it will burn fuels which are much lower in cost than the highly volatile and refined fuels ordinarily used in automotive engines. It has been claimed for this engine that its specific consumption is only half as great and that it uses fuels costing only one-seventh as much as gasoline, so the fuel cost would be in the ratio of 14 to 1. This comparison probably is slightly overdrawn, but it is a fact that the Diesel engine cuts fuel costs to a small fraction of that with gasoline in carburetor engines.

There are three general types of Diesel engine fuel, namely, crude petroleum oil, tar oil and crude vegetable oil. In this country crude petroleum oil would be the natural Diesel engine fuel to use. In this connection the ignition temperature of the fuel is of importance, as the higher this temperature, the higher the compression pressure must be carried to insure ignition under all conditions. Tar oils generally have the highest ignition temperatures, and in stationary practice it is customary to use a small quantity of a so-called ignition oil which is injected ahead of the regular fuel charge to initiate the combustion. Ignition temperature tests made by Mirrless, Bickerton & Day, a British firm of Diesel engine builders, with various fuels in oxygen, gave the following results in degrees F.; Gasoline (two grades) 518° and 522°; kerosene, 486;° gas oil, 490°; fuel oil, 501°; shale oil, 488°; tar oil, 879°; coke oven tar, 922°; water gas tar, 866°; blast furnace tar, 930°; whale oil, 524°; naphthalene 756°; alcohol, 744°.

If only petroleum fuels are to be used the compression need not exceed 400 lbs. per sq. in. to insure ignition.

Not the least of the advantages of the Diesel engine is its clean or complete combustion. Crankcase oil dilution and carbon deposits of any consequence do not occur with it, and the exhaust is at least substantially free from carbon monoxide under all conditions of load.

Characteristics of the Engine—The two outstanding mechanical features of the Diesel engine are that it compresses pure air, instead of an explosive mixture of air and fuel vapor, and that ignition is effected by the heat of compression. The same as carburetor engines, Diesel engines are built in both the four-stroke and the two-stroke type. Some of the disadvantages of the two-stroke cycle carburetor engine are non-existant with the Diesel; for instance, since only air is taken into the cylinder during the inlet period, there is no possibility of any fuel being lost through the exhaust port during the charging period; and, besides, the gases of combustion can be swept from the cylinder by a scavenging charge of air, so that the cylinder can be completely filled (and even supercharged, if desired). This advantage, together with its simplicity of construction, has led to extensive use of the two-stroke engine in large marine and stationary installations in recent years. But automotive work calls for high speed operation, and the fact that the inlet and exhaust periods are so much shorter in the two-stroke engine, makes it less suitable for high speed work, and nearly all development work on Diesel engines with automotive applications in view seems to have been done on the four-stroke type.

The chief elements of the Diesel four-stroke engine are the same as those of the corresponding Otto engine, and since the reader is already familiar with the latter, it is only necessary to explain the differences between the two.

For ease in mounting on the chassis, etc., the automotive Diesel engine is generally made in the vertical form. The valve-in-head type of construction is used exclusively with these engines. In the cylinder head there are two fairly large sized poppet valves, for air admission and for the exhaust. Air is often taken in through a muffler consisting of a length of pipe with numerous narrow slits in it. During the second stroke the air in the cylinder is compressed, the volumetric compression ratio being generally 14 to 1 or a little greater. During compression the pressure increases in accordance with the equation

$$P_1 = (V/V_1)^{\gamma} P.$$

In Chapter III we figured with a value of 1.30 for γ, but in this case, owing to the fact that compression is carried so much further, we can figure with a value of 1.35; the initial pressure P may be assumed to be 14 lbs. per sq. in., and the volumetric compression ratio V/V_1 being 14, we get for the final pressure

$$P_1 = 14^{1.35} \times 14 = 495 \text{ lbs. per sq. in., abs.}$$

or about 480 lbs. gauge. If air taken in at atmospheric temperature is compressed adiabatically in this proportion its temperature is raised to about 1000° F. At the end of the compression stroke fuel is injected or sprayed into this highly heated air; it ignites spontaneously and burns at the rate at which it is injected, provided, of course, it is thoroughly atomized and well distributed throughout the air charge.

The reason for compressing air only, instead of a combustible mixture, is that there is then no possibility of preignition, and the compression can be carried as high as desired, with consequent increase in thermal efficiency and the further advantage that ignition is spontaneous and no ignition equipment is required.

Combustion at Constant Pressure—Another peculiarity of the Diesel cycle may be referred to here. With the Otto cycle all of the fuel is present in the cylinder when ignition takes place, and although combustion is not instantaneous, if the mixture is of the proper proportion, it takes place very rapidly while the charge is compressed to substantially its minimum volume. The heat of the fuel is therefore liberated and appears in the charge as sensible heat while the latter is at constant volume—the volume of the compression space.

In the Diesel engine the fuel is injected during a fraction of the down stroke varying with the load on the engine, the aim being to so control the rate of injection that the addition of heat due to the combustion of the fuel will maintain a constant pressure, where otherwise the pressure would drop because of the expansion. Thus in large Diesel engines the air is compressed in the cylinders to about 500 lbs. per sq. in. and this pressure is maintained as long as the fuel injection lasts, after which the pressure drops in accordance with an exponential curve, the same as in an Otto engine. A typical diagram from a large Diesel engine is shown in Fig. 357.

In a high-speed engine the combustion takes place under somewhat different conditions, and pressure variations also are different. There is undoubtedly a certain fixed time lag between the instant the pump begins to deliver fuel and the instant the first fuel molecules are burned, which is independent of the speed of the engine. Since it is the combustion that should always take place at the same point of the cycle, in a high-speed engine the pump must be given a certain lead or advance, the same as the ignition spark in a high-speed carburetor engine. But when the fuel is thus injected before the piston reaches the top of the stroke, a phenomenon similar to that in a carburetor engine takes place; that is, a large part of the fuel may burn almost instantaneously while the piston is at or near the top of the stroke and thus give rise to a large increase in pressure. This is what actually occurs in small high-speed Diesel engines. Dr. Riehm of the M. A. N., whose truck engine will be referred to later, states that with solid injection a combination of a constant pressure and a constant volume diagram is obtained, and with a compression pressure of 375 lbs. the pressure will rise to 600 lbs. during combustion.

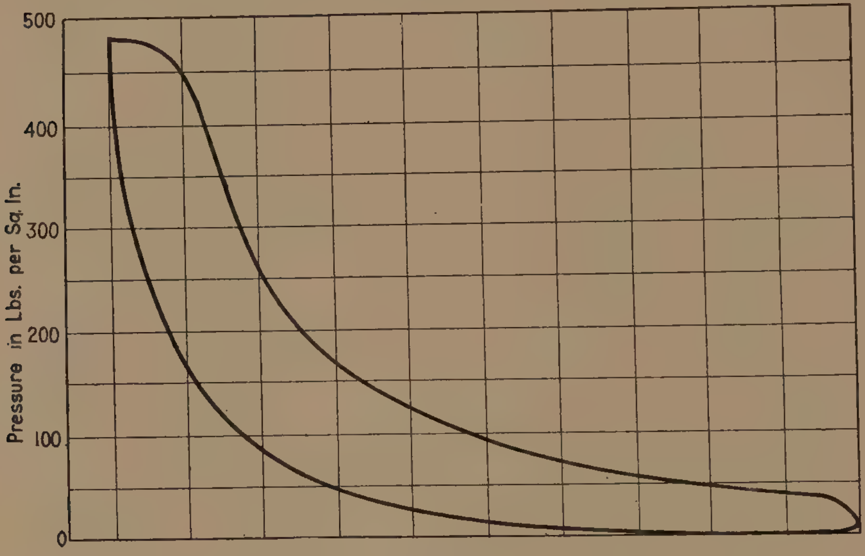


FIG. 357.—DIAGRAM FROM LARGE DIESEL ENGINE WITH AIR INJECTION.

Theoretical Efficiency of Diesel Cycle—In Fig. 358 is shown a theoretical cylinder diagram of a Diesel engine. Using the same notation as in the discussion of the Otto cycle diagram (Chapter III), the air is taken into the cylinder at a pressure P_0 and at a temperature T_0, both in absolute measure. The air is then compressed in the cylinder adiabatically until the pressure is P_1 and the corresponding temperature T_1. Then heat is added while the air expands, the pressure remaining the same but the temperature rising to T_2. Next the burning gases expand adiabatically until at the end of the stroke the pressure reaches P_3 and the temperature T_3. The gases are then discharged from the cylinder and the temperature becomes the same as at the beginning of the cycle, viz., T_0.

Now let us assume that r is the ratio of compression of the air and R the ratio in which the cylinder volume increases while fuel is being injected. Since heat is supplied to the charge at constant pressure, the amount supplied to unit mass is evidently $c_p(T_2 - T_1)$, while the amount of heat rejected at the

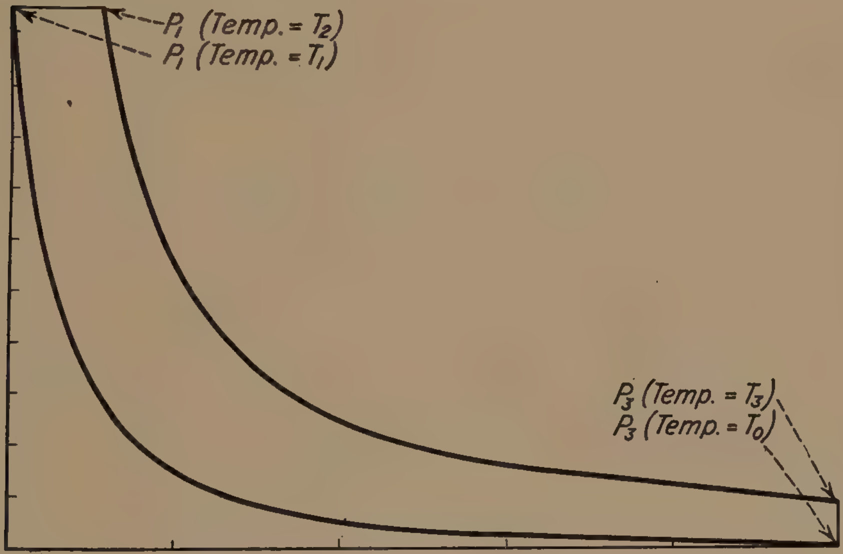


FIG. 358.—THEORETICAL DIAGRAM OF DIESEL ENGINE.

end of the expansion stroke is $c_v(T_3 - T_0)$. The efficiency of the cycle is

$$\frac{\text{Heat supplied} - \text{Heat rejected}}{\text{Heat supplied}},$$

and we therefore have

$$\eta = \frac{c_p(T_2 - T_1) - c_v(T_3 - T_0)}{c_p(T_2 - T_1)}$$

$$= 1\frac{c_v(T_3 - T_0)}{c_p(T_2 - T_1)} = 1 - \frac{T_3 - T_0}{\gamma(T_2 - T_1)}.$$

According to equation (5), Chapter II, in adiabatic compression and expansion,

$$TV^{\gamma-1} = \text{Constant};$$

hence, since the air is compressed adiabatically,

$$T_0V_0^{\gamma-1} = T_1V_1^{\gamma-1},$$

and

$$T_1 = \frac{V_0^{\gamma-1}}{V_1}T_0 = r^{\gamma-1}T_0.$$

The expansion takes place at constant pressure, under which condition the volume varies in direct proportion to the absolute temperature. The expansion ratio during fuel injection, also referred to as the cut-off ratio, being R, it follows that

$$T_2 = RT_1 = Rr^{\gamma-1}T_0.$$

At the beginning of the adiabatic expansion the charge occupies a volume R and at the end, a volume r; hence, making use again of equation (5) we get

$$T_2R^{\gamma-1} = T_3r^{\gamma-1},$$

$$T_3 = \frac{R^{\gamma-1}}{r}T_2 = \frac{R^{\gamma-1}}{r} \times R \times T_1 = \frac{R^{\gamma}}{r^{\gamma-1}}T_1 = R^{\gamma}T_0$$

We now have expressions for T_1, T_2 and T_3 in terms of T_0 and can substitute these in the equation for η. This gives

$$\eta = 1 - \frac{R^{\gamma}T_0 - T_0}{\gamma(Rr^{y-1}T_0 - r^{y-1}T_0)}$$

$$= 1 - \frac{R^{y} - 1}{\gamma(Rr^{y-1} - r^{y-1})}$$

$$= 1 - \frac{R^{y} - 1}{\gamma r^{y-1}(R - 1)}.$$

To show how the efficiency varies with the cut-off, assume that the compression ratio r is 14; then for values of the cut-off ratio R of 1.25, 1.5, 2 and 2.5, corresponding to fuel injection during about 2, 3.9, 7.7 and 11.5 per cent of the stroke, respectively, the theoretical efficiencies would be 63.6, 62.0, 59.2 and 56.8 per cent, respectively. Thus the theoretical efficiency increases as the load on the engine decreases. As compared with the above figures, an Otto engine with the usual compression ratio of 4.5 has a theoretical efficiency (calculated on the same basis) of 45.2 per cent, which is the same for all loads, as the expansion ratio does not vary. Theoretical considerations therefore indicate that the thermal efficiency of the Diesel engine is about 40 per cent greater than that of the Otto engine.

Fuel Injection—The injection of fuel in exactly measured quantity and at an accurately regulated rate presents one of the most difficult problems in connection with the Diesel engine, particularly with automotive type engines, because of the necessarily small size of their cylinders. With the heavier hydrocarbons usually employed in Diesel engines the volumetric proportion of the liquid fuel to the air necessary to burn it is about 10,000 to 1, hence the bore and stroke of the fuel pump must be less than 1/20 of the corresponding dimensions of the engine cylinder. Moreover, it is not sufficient that the proper amount of fuel be introduced into the cylinder at the proper time—it must also be thoroughly atomized and uniformly distributed, so that the combustion may be practically completed when the cut-off occurs.

In most large Diesel engines the fuel is blown into the combustion chamber by air under a pressure of about 1000 lbs. per sq. in. This method results in very thorough atomization, but there are two objections to it. The first is that as the injection air expands in the less highly compressed air in the compression chamber, it has a chilling effect, and at low speeds, when more heat is lost through the cylinder walls, this may interfere with ignition.

The second (and from the automotive standpoint probably the most serious) objection to the system is that it involves considerable complications. It is necessary, in the first place, to have an air compressor capable of compressing air to 1000 lbs. per sq. in. For the highest efficiency in large installations this is sometimes done in four stages, the air being cooled after each stage. For smaller engines a two-stage compressor suffices, with an inter-cooler between the low-pressure and high-pressure cylinders. The compressor delivers through an after-cooler and oil separator into an air bottle which serves to even out the pressure and to provide a source of injection air immediately after starting. The ‘‘after cooler’’ and oil separator are safety devices, as without them an explosive mixture may form and be set off by the heat of compression, exploding the air bottle or its connecting pipe.

Large Diesel engines are always started by means of compressed air, which is admitted to their working cylinders through a special starting valve. This is a poppet valve in the cylinder head adjacent to the air inlet and exhaust valves, but of smaller size. It is operated by a cam on the camshaft, and an interlocking mechanism is provided which prevents the opening of the fuel injection valve when the starting valve is being operated, and vice versa.

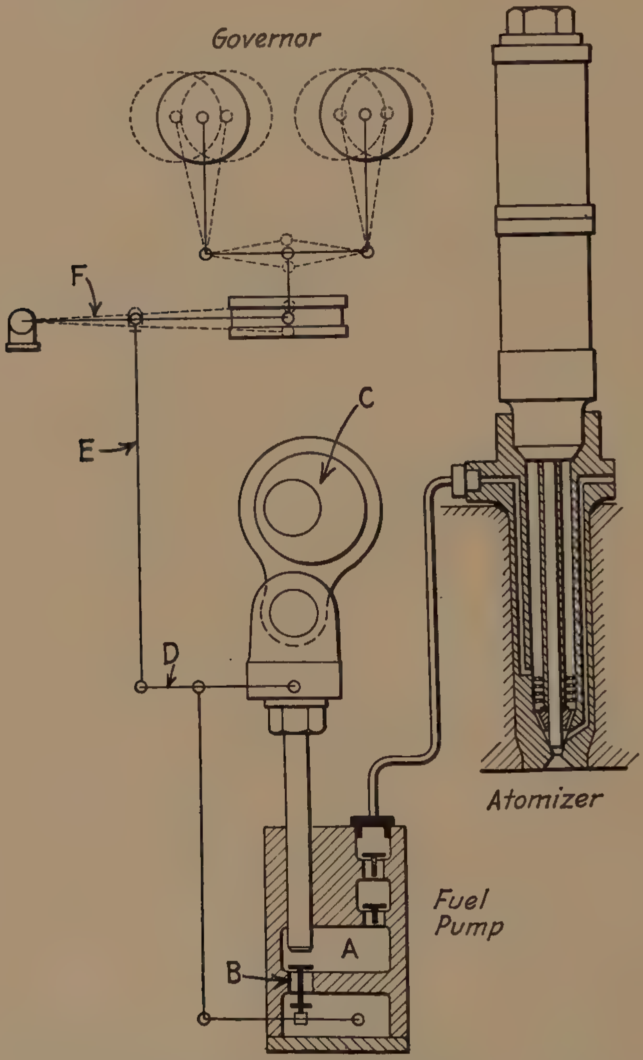


Fig. 359.—Diagrammatic Representation of Fuel Pump, Atomizer and Governor.

Fuel Pump—Fuel is delivered to the atomizer by a piston type of fuel pump which is generally operated off the camshaft by an eccentric, as illustrated more or less diagrammatically in Fig. 359. Fuel flows to chamber A at the lower end of the pump barrel by gravity from an overhead supply tank. The stroke of the pump is invariable, and the delivery is controlled by holding the suction valve B open during a smaller or greater part of the return stroke. The suction valve is lifted off its seat by means of a linkage connecting to the eccentric C which reciprocates the pump plunger. In this linkage there is a lever D having a floating fulcrum, the fulcrum point being connected to the engine governor by link E and lever F. If the speed exceeds that for which the governor is set, the fulcrum point of the lever is raised, with the result that suction valve B remains open during a greater part of the return stroke of the pump plunger and more of the fuel drawn into the pump barrel during the suction stroke is forced back through the valve during its return stroke, hence less is delivered to the atomizer. The engine is stopped by manually holding the fuel valve off its seat, and provisions are also made for varying the speed of the engine, by moving either the fulcrum of lever F or the support of the governor spring.

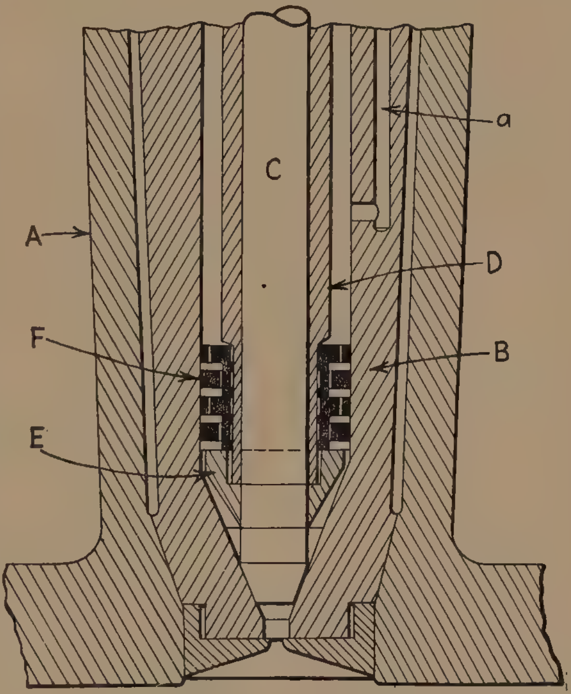


Fig. 360.—Air Injector Atomizer.

Atomizer. — The atomizer is generally located at the center of the cylinder head, between the air and exhaust valves. It consists of a cage set into a drill hole in the cylinder head which hole has a taper seat near its lower end. The cage is secured to the cylinder head by means of a flange on it, and studs and nuts. Above the cylinder head it is provided with connections for the injection air and the fuel.

A typical design of atomizer used with air injection is shown in Fig. 360. A is the wall of the cylinder head into which is set the shell B of the atomizer. In the center of the shell is located the atomizer needle C which is surrounded by a guide D for the greater part of its length. This valve guide is supported at its lower end by the cone E with a considerable number of fine fuel slots in its outer surface. Above the cone and upon a reduced portion of the guide D there are a number of perforated washers F separated by other washers of smaller diameter. The perforations in adjacent washers—there are as many as 20 in each washer—are staggered. The atomizer needle is seated in the lower part of the shell or cage and over the lower end of the latter is screwed an atomizer cap.

The space between the atomizer shell and the valve guide is at all times filled with injection air at high pressure, the needle valve being normally held to its seat by a strong spring (not shown). Fuel is forced into the atomizer by the pump through the passage *a* in the shell. When the atomizer needle is lifted, about 5° ahead of dead center, the injection air forces the fuel charge through the perforations in the washers, the slots in the cone, and the orifice in the atomizer cap, and delivers it in a fine spray into the charge of air in the compression chamber, where it is spontaneously ignited and burned.

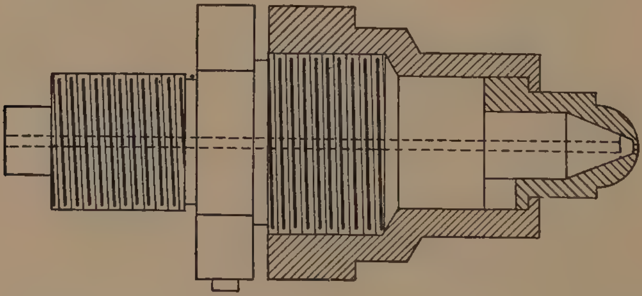


FIG. 361.—SOLID INJECTION NOZZLE (M. A. N.).

Solid Injection—Another method of fuel delivery is known as solid injection and consists in forcing the fuel into the combustion chamber directly by the pressure of the pump, through a nozzle which has a tendency to spray it finely. It is generally admitted that the atomization and the distribution of the fuel in the air obtainable with solid injection are not as good as with air injection, and that the combustion with the latter system therefore is not quite as efficient, but the simplicity of the solid injection system has brought it into great favor with Diesel engine designers of late years. A nozzle for a solid injection fuel valve is shown in Fig. 361.

Shape of Combustion Chamber—With solid injection the best shape of the combustion chamber is a problem of much importance. With air injection the fuel valve is generally arranged in the axis of the cylinder and the compression space is made flat. This does not give good results with solid injection as the fuel stream on striking the relatively cold piston head seems to adhere to it, and the distribution suffers. It has therefore become customary to make the piston head almost hemispherical when the fuel valve is in the cylinder axis. Another plan that has been adopted consists in using two fuel valves, on opposite sides of the cylinder, and with this arrangement it has been found advisable to have the axes of the two offset, rather than in line. The latter arrangement was expected to improve atomization and distribution, but the opposite effect was obtained. Much experimentation on the best shape of combustion chamber and best location and direction of nozzles with solid injection is now going on.

Weight of Engines—One of the principal problems in connection with the adaptation of Diesel engines to automotive work is that of weight reduction. The lowest weight previously attained in this class of engine for other purposes (submarines) is about 50 lbs. per hp., and this must be at least halved.

When looking at a conventional type of Diesel engine one is struck with the robustness, the ponderousness of its construction. The factor of safety allowed seems to be unnecessarily high when it is considered that the normal working pressure in a Diesel engine is only about 50 per cent greater than that of an Otto engine. However, a good many engineers who have been in the Diesel engine business for a long time know of explosions which have occurred, in which the cylinder head was torn from the engine or even the whole cylinder parted from the crankcase.

The explanation of the excessive pressures which caused these accidents is that when the piston is at the top end of the stroke, the air in the cylinder is compressed to approximately 500 lbs. per sq. in. or more than 5 times as high as in an Otto type engine, and if by any mischance fuel should have gotten into the cylinder prematurely, in sufficient quantity to require all of this air for its combustion, a pressure would be developed at least 5 times as high as the maximum pressure in an Otto type engine. Such pressures must be provided for in the design, and for this reason, although the mean effective pressure of the Diesel engine is only about 100 lbs. per sq. in., all of its working parts are generally given sufficient strength to withstand the strains due to a maximum gas pressure of 2000 lbs. per sq. in.

Temperature Curves—In large Diesel engines the maximum temperatures reached by the gases and the temperatures throughout the cycle are of great importance, as it is difficult to cool the large cylinders and pistons so as to make lubrication possible and prevent injury to the parts by thermal strains, which often cause cracking. In automotive engines these problems do not occur and the curves of Diesel engine cylinder wall temperatures in Fig. 362 are shown here chiefly because of the light they throw on the phenomenae occurring inside a Diesel cylinder. These curves were obtained by Eichelberg in 1923. Curve A represents the variations in temperature which would be expected on theoretical grounds, while curves B show the temperature variations at full, ¾, ½ and ¼ load. The maximum temperature reached at full load is about 2850° F.

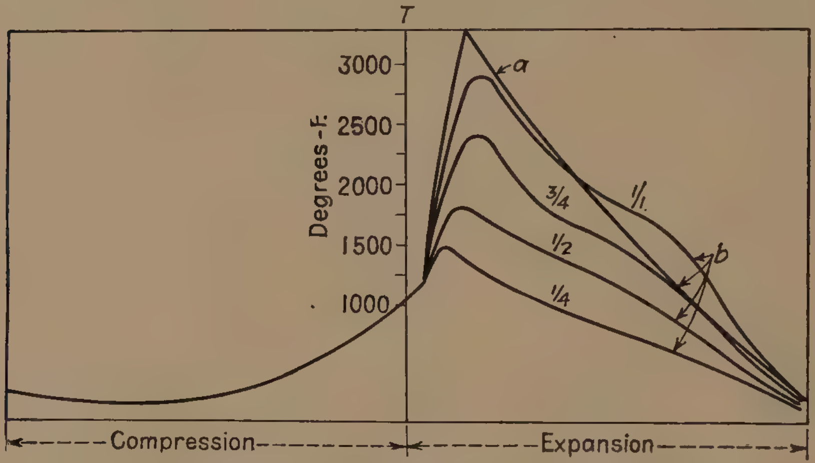


FIG. 362.—TEMPERATURE CYCLES IN DIESEL ENGINE.

Auxiliary Ignition Chamber—In automotive work, with cylinders of small dimensions and, consequently, relatively large cooling area, and where low engine speeds are often required, the problem of spontaneous ignition offers additional difficulties, and one solution lies in the use of an auxiliary ignition chamber. This device is made use of on the Benz engine, the cylinder head of which is illustrated in Fig. 363. The ignition chamber is located between the air valve and the exhaust valve. The fuel injection valve is screwed into the top wall of this chamber, which latter communicates with the compression chamber through a necked passage. Into this passage is screwed a device known as a "burner"—a sort of nozzle provided with a number of small orifices at its lower end. The "burner" is located between the two valve chambers where there is no water jacket, and its central section is relieved on the circumference so as to retard the heat flow from it into the head casting.

During the compression stroke of the piston, air is forced into the auxiliary chamber through the orifices in the burner, and at about 8° of crank motion ahead of dead center, the fuel injection valve is opened and fuel sprayed through the auxiliary chamber onto the walls of the "burner." Ignition is effected by the high temperature of the "burner" wall combined with the heat of compression. A portion of the fuel is burned in the auxiliary chamber, and the resulting pressure in this chamber forces the remainder of the fuel through the orifices of the "burner" into the compression chamber.

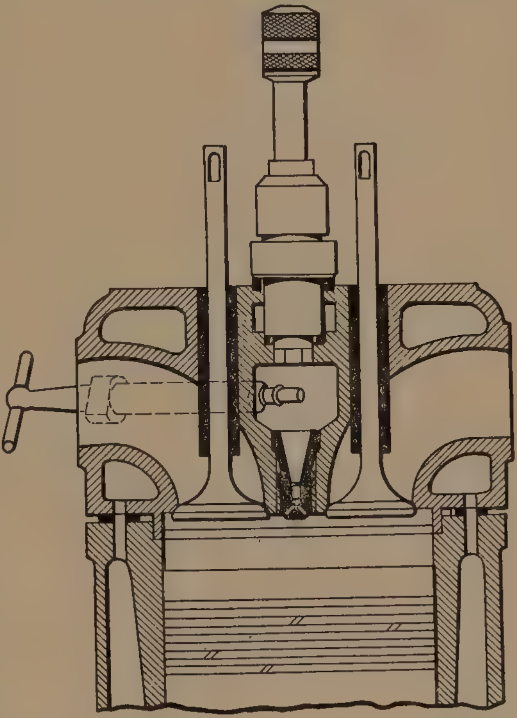


FIG. 363.—CYLINDER HEAD OF BENZ-DIESEL TRACTOR ENGINE.

Benz Atomizer.—The atomizer used on the Benz engine is illustrated in Fig. 364. It sets in a central bored recess in the cylinder head and is secured by a flange and bolts or studs. Fuel from the pump enters it through the pipe connection at the right. The atomizer needle is held to its seat by a coiled spring; its stem enters a guide which is blind at the upper end, and when the pressure on the oil is increased by the pump action, the air above the stem is compressed and the valve raised off its seat. On top of the atomizer there is an air relief valve or vent, through which air (which would interfere with the metering) can be allowed to escape from the atomizer chamber. This is a conical valve which is normally screwed down tight on its seat. It is opened by means of its knurled head.

Fuel Pump—The fuel pump, illustrated in Fig. 365, is a plain plunger pump without packing. The plunger is positively forced into the barrel by cam action and returned by a coiled spring. Suction and discharge valves of conical type are used, the former being closed by the action of gravity and the pressure on the fuel in the pump, while the discharge valve is held to its seat by a spring. In this design the stroke of the plunger rather than the closing point of the inlet valve is varied to take care of changes in load and to effect changes in speed.

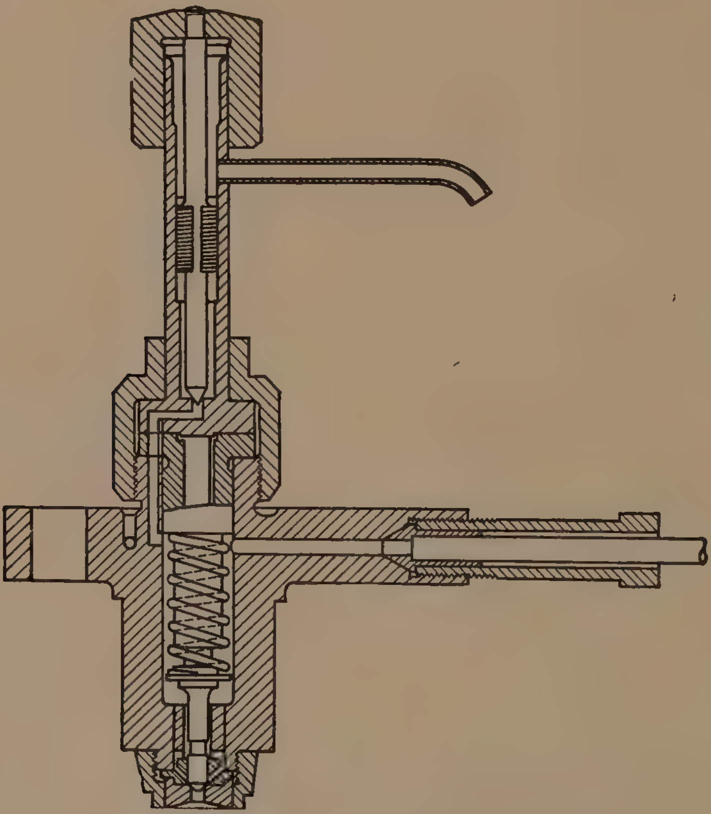


FIG. 364.—INJECTOR VALVE OF BENZ ENGINE.

Referring to the sectional view of the pump and its operating mechanism, the pump plunger A is acted on by the tappet B, which latter is properly guided and located over an adjustable stop. Tappet B is acted on by the cam follower lever provided with a roller on which the cam E acts. This lever has a floating fulcrum on a link connecting to forked lever G, to which latter the governor rod H is also connected. The floating fulcrum pin also carries a shoe which slides on a guide surface formed on the housing and takes up the horizontal component of the thrust of the cam on the roller. As the governor rod moves upward, the fulcrum of the cam lever is moved away from the axis of the cam. The height of the free end of the cam lever which corresponds to the beginning of the pump stroke is fixed by means of the adjusting screw. Raising the movable fulcrum reduces the stroke of the plunger, and vice versa. The thrust or reaction shoe I is held against its guide surface by a flat spring J. By means of a hand lever L the cam lever can be held out of contact with the cam, as a result of which the fuel feed is interrupted and the engine stopped.

A sectional view of the governor and control linkage is shown in Fig. 366. The governor weights AA act through

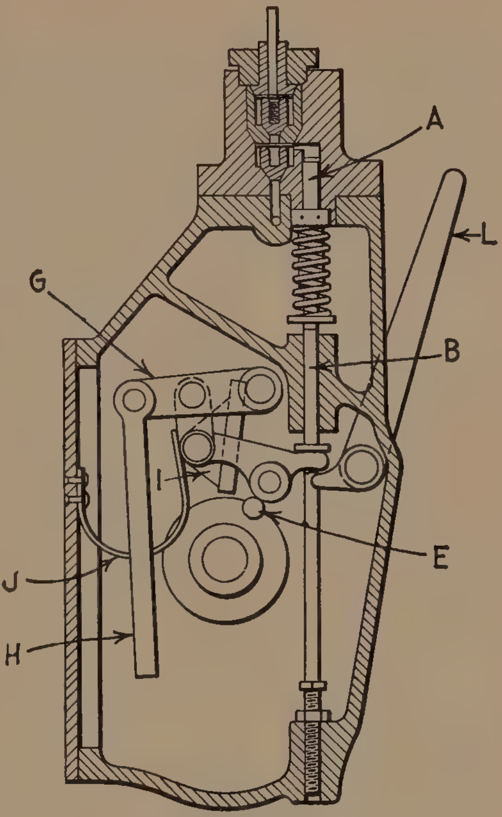


Fig. 365.—Fuel Pump of Benz Engine.

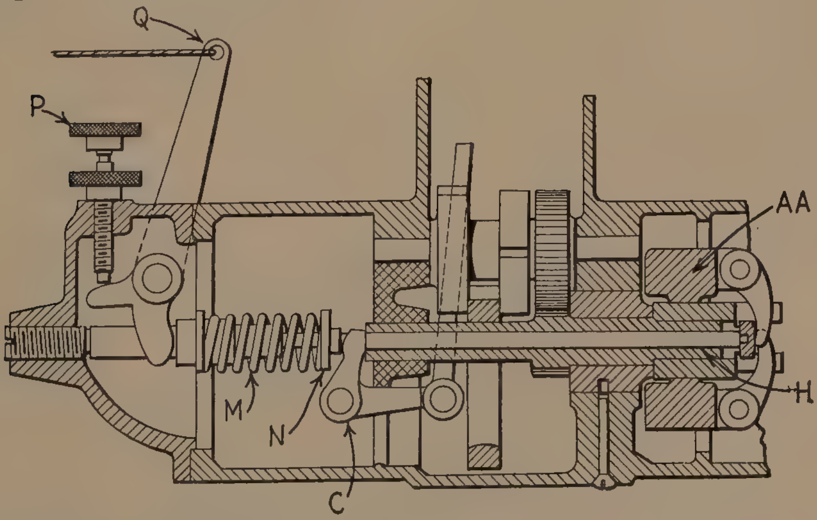


Fig. 366.—Governor and Control Linkage of Benz Engine.

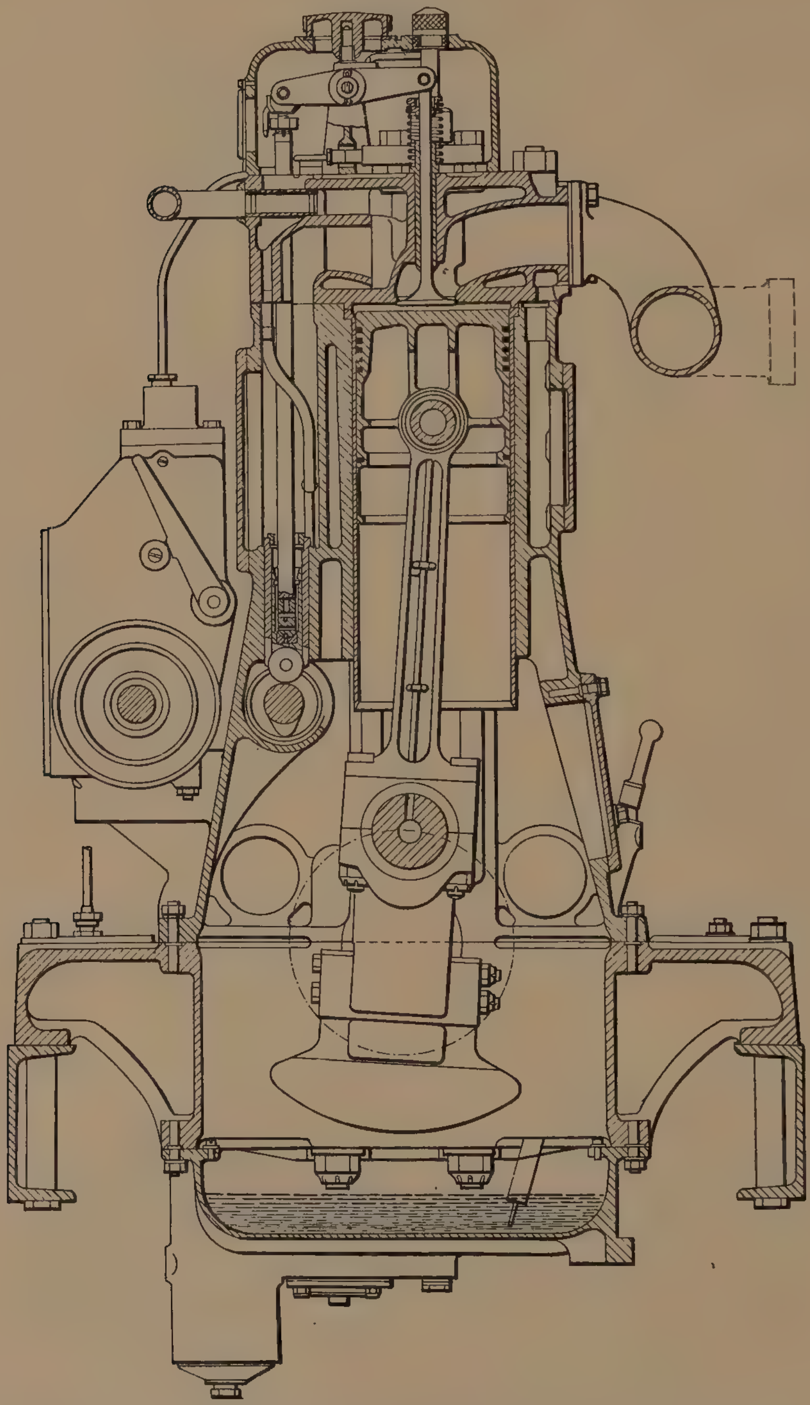


Fig. 367.—Cross-section through Benz-Diesel Engine.

the usual inwardly extending arms on a rod H passing through the hollow governor shaft. The other end of rod presses against a bell crank C, to the other arm of which the governor rod is connected. In line with rod H is located the governor spring M, the spring rest N of which rests on a bell crank C. The pressure of the governor spring can be varied by means of adjusting screw P, which permits of altering the governed speed of the engine; it can also be varied by pulling on the cord connected to the lever Q, which attaches to the speed control lever at the driver's seat.

Characteristics of Benz Engine—The Benz engine has been built in a four-cylinder type for trucks and a two-cylinder type for tractor service, but the latter seems to be the later design and to incorporate a number of improvements over the four-cylinder model. Its two cylinders are of 5.32 in. bore by 7.88 in. stroke. The cylinders and upper section of the crankcase are a single aluminum casting into which cast iron liners are pressed. Each cylinder has a separate detachable head, and on top of this there is a housing for the valve mechanism. The two cranks are set at 180° with each other and the crankshaft is counter-weighted. A cross-section of the engine is shown in Fig. 367.

Method of Starting—This engine is started by hand. First a starting cam is brought into action by shifting the camshaft axially, to keep the exhaust valves open during the first part of the compression stroke and reduce the compression to 170 lbs. per sq. in. This results in an air temperature that will ignite some of the fuels of low ignition point. However, to make starting from cold positive even on the less ignitable fuels, an auxiliary igniter is provided, consisting of a 1¼ × 1½ in. piece of blotting paper formed into a 5/16 in. roll and inserted into a holder. This blotting paper is saturated with nitrate of potash and burns faintly when lighted with a match. After this "cartridge" has been inserted into the ignition chamber the engine is cranked over under reduced compression. Fuel for the first charge is furnished by the roll of blotting paper. As soon as an ignition has been obtained the setting may be changed to that of normal operation and the engine will immediately carry its full torque. Lubrication of this engine is by a force-feed system similar to that used on automobile engines.

Engine Performance—Performance curves of the Benz-Diesel engine are shown in Fig. 368. The maximum output is 32 hp. at a little under 800 r.p.m., corresponding to a b.m.e.p. of 112.5 lbs. per sq. in. Horse power and fuel consumption are plotted against engine torque. It will be seen that the governor keeps the engine within speed limits of 780 and 840 r.p.m. The driver, by means of his speed-control lever acting on the governor spring, can reduce the speed to 250 r.p.m. at any load, without jeopardizing the certainty of ignition, it is claimed.

The fuel consumption (tar oil) reaches a minimum of 0.45 lb. per hp.-hr., and does not increase materially down to half load. For smaller loads it is considerably greater, owing to almost constant friction and due to the driving of auxiliaries.

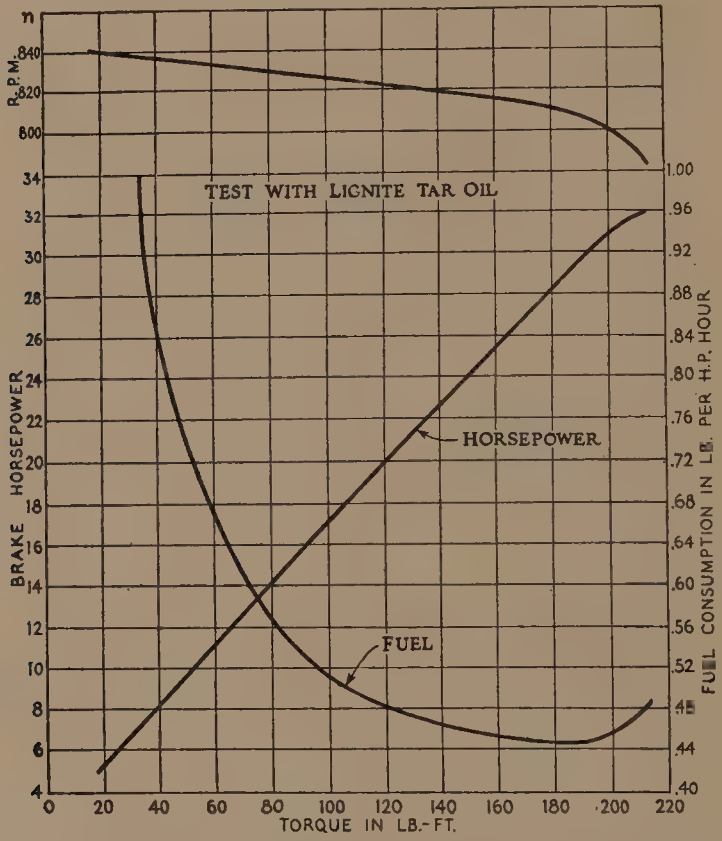


FIG. 368.—PERFORMANCE CURVES OF BENZ-DIESEL ENGINE.

M. A. N. Truck Engine—Figure 369 is a sectional view of a four-cylinder truck engine which has been in production by the Maschinenfabrik Augsburg-Nurnberg since 1924. In its structural form it hardly differs from a carburetor type of truck engine. Its four cylinders are cast in a single block of cast iron, while the crankcase is of the barrel type and cast of aluminum. Solid injection is used, and there are two injector nozzles, on opposite sides of the flat compression

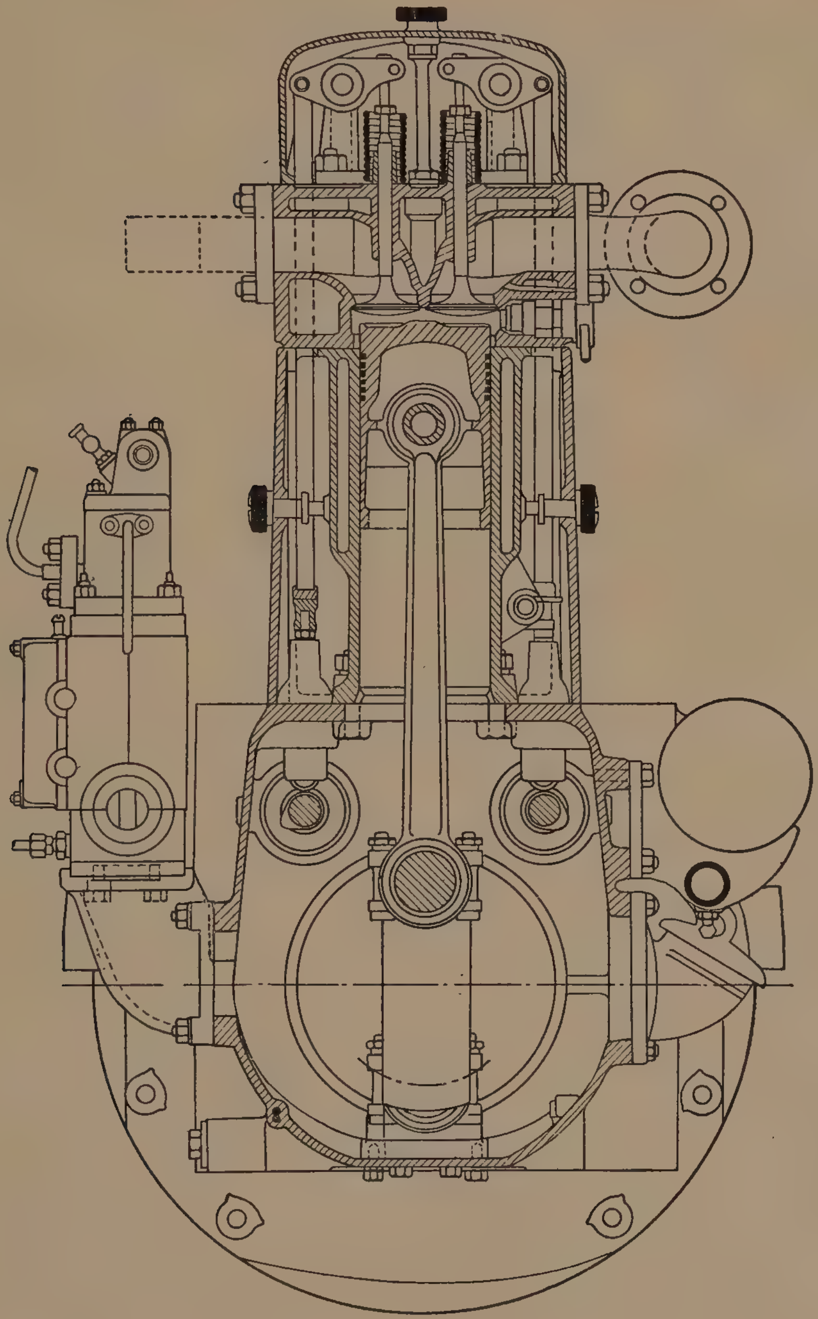


FIG. 369.—CROSS-SECTION THROUGH M. A. N. FOUR-CYLINDER TRUCK ENGINE.

chamber. Fuel is delivered to these nozzles by a cam-operated fuel pump mounted on one side of the engine. In fact, this pump occupies the place of the usual magneto, while the fuel nozzles take the place of the spark plugs.

An unusual feature is that there is no movable part in the fuel line beyond the delivery valve of the pump, which is located in the distributor head. The fuel nozzle (Fig. 361) has only three parts, the nozzle body, nozzle plate and nozzle cap, and can be readily dismounted. Having the nozzles on the sides of the engine makes it possible to use larger valves in the head, which is desirable for a high-speed engine.

This engine has a bore of 4½ and a stroke of 7⅛ in. and develops 32 hp. at 600 r.p.m., 50 hp. at 1000 r.p.m. and 54 hp. at 1200 r.p.m. The weight is about 1100 lbs. or slightly over 20 lbs. per hp.

Air inlet and exhaust valves are operated from separate camshafts in the crankcase, and great pains seem to have been taken to produce a neat, symmetrical design. Starting can be effected either by hand or electric starter. At 800-1000 r.p.m. the consumption of gas oil is around 0.44 lb. per hp.-hr. between 35 and 50 hp.; at 1200 r.p.m. it is somewhat higher. Road tests are said to have shown a saving in fuel of 25 per cent over a carburetor engine of equal power. The b.m.e.p. drops from 94 lbs. per sq. in. at 600 r.p.m. to 82 lbs. at 1100 r.p.m.

Hindlmeier Engine—Another attempt to adapt the high pressure principle of operation to small units is represented by the Hindlmeier engine, which was brought out in Austria about 1920 and for the manufacture of which licenses were issued in different countries, including the U. S. While the Austrian licensees some years ago produced a four-cylinder truck engine production seems to be confined chiefly to small single and double cylinder stationary units.

To facilitate starting and to minimize the difficulties ordinarily associated with the use of very high pressures, only a small portion of the air of combustion is highly compressed. This air, at about 700 lbs. per sq. in., is used for both injection and ignition. It is compressed in a single stage in a cylinder parallel to the working cylinder, containing a piston operated from the half-speed shaft. This air passes directly from the compressor cylinder to the injection valve, and air storage is thus avoided.

Compression in the working cylinder is carried to 400 lbs. per sq. in. In multi-cylinder engines a separate compressor cylinder is required for each working cylinder. This, of course, is a handicap and probably explains why the manufacturers confine themselves to single and double cylinder types. The truck engine referred to operated at a normal speed of 1150 r.p.m.

Compound Diesel—Elmer A. Sperry of Brooklyn, N. Y., has done some work with compound engines working on the Diesel principle and exhibited an engine-generator set at the automobile show in New York some years ago. Sperry reasons that compounding offers two important advantages. By looking at a Diesel engine indicator diagram it is seen that exceedingly high pressures are produced in such an engine, but that these high pressures are maintained for only a small fraction of the stroke. If the expansion were performed in two cylinders, only the small high-pressure cylinder would need to be built to withstand these high pressures, while the low-pressure cylinder could be built with light walls and the total weight thus materially reduced. A further advantage would be that, whereas in the ordinary Diesel engine with a compression ratio of 14 or 15 to 1, the compression space is exceedingly flat and therefore has a large cooling area in proportion to its volume, with two-stage expansion the expansion ratio in the high-pressure cylinder need not be more than 2.5 or 3 to 1, and this would give a much more favorable compression chamber shape.

Hvid Engine—In the Hvid engine the fuel and a small quantity of air are drawn into a cast iron cup directly under the cylinder head during the inlet stroke by the suction in the cylinder. The amount of fuel is determined by a needle valve, the setting of which can be changed by hand while running. The cast iron cup communicates with the combustion chamber through a fine opening.

During the compression stroke air enters the cup through the small opening, and when the pressure in the cup reaches a certain value, ignition takes place, a small portion of the fuel burns and the pressure in the cup rises, whereby the remainder of the fuel is forced out through the small orifice and energetically atomized. The flame being communicated to the combustion chamber, the rest of the fuel burns. The cup, of course, forms a hot surface which helps ignition.

Surface Ignition Engines—An intermediate type between the Diesel and the carburetor engine is that in which the working cylinder communicates with a chamber whose walls are not water-cooled but are protected against cooling effects by an air jacket. This principle was originated by Akroyd in England in 1892. In the usual form the hot chamber or bulb is located on the cylinder head, and for starting can be heated by a torch. Fuel is injected into this chamber

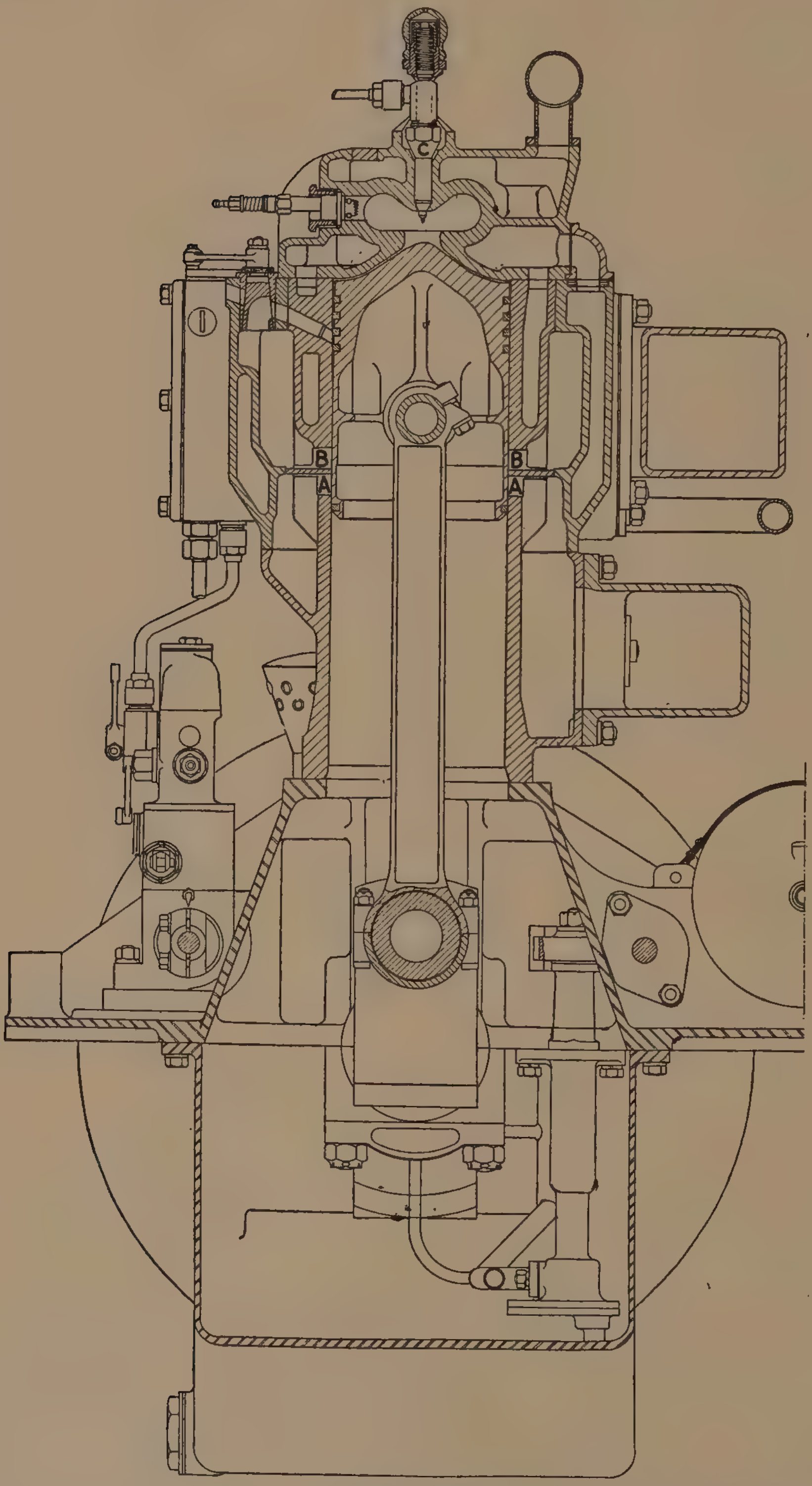


FIG. 370.—CROSS-SECTION OF TARTRAIS-PEUGEOT TWO-STROKE HEAVY OIL ENGINE.

at the same time that air, heated by the compression, enters it from the cylinder, and the fuel is ignited spontaneously.

Oil engines of this type have been widely used for stationary and marine work. One design known as the Tartrais was developed by the Peuegot firm in France for automotive purposes. On top of the cylinder there was an annular, air-jacketed chamber into which the air drawn into the cylinder was compressed. However, in the latest design, shown in section in Fig. 370, the combustion chamber is water-jacketed. This engine works on the two-stroke cycle and employs a special scavenging pump. There are two cylinders side by side, with cranks at 180°, and a counter-weighted crankshaft. From the sectional view it will be seen that when the piston is in the top dead center position there is practically no clearance in the cylinder, and the rush of compressed air into the ignition chamber toward the end of the compression stroke must create a very strong degree of turbulence.

Fuel is injected into the ignition chamber from the top and is ignited by the heat of the compression, which is carried at about 375 lbs. per sq. in., the volumetric compression ratio being 12 to 1. Starting is effected by means of an electrically heated coil supported by a plug screwed into the wall of the ignition chamber.

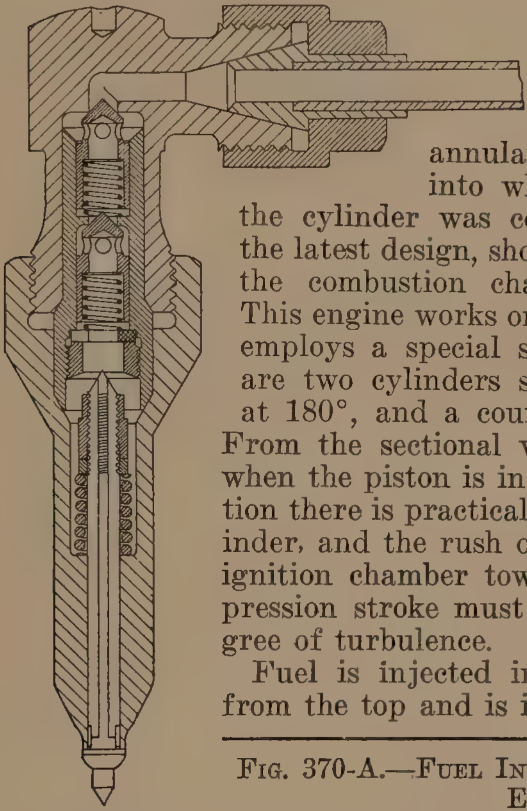


FIG. 370-A.—FUEL INJECTOR VALVE OF TARTRAIS ENGINE.

CHAPTER XXIII

Water Cooling

The heat absorbed by the jacket water from the burning charge must be transferred to the atmosphere as quickly as it is taken up. To this end the jacket water is circulated through a radiator, which generally is located forward of the engine, where, when the car is in motion, it receives the full benefit of the draft created by that motion, but which in a few cases is located back of the engine in the position usually occupied by the cowl, the advantage of this latter arrangement being that it gives better access to the engine. The cooling water must be kept in constant circulation. This circulation can be induced either by means of a pump or by so arranging the circuit that the expansion of the water by the heat absorbed in the jacket and its consequent tendency to rise, causes it to circulate with sufficient rapidity. The first system, known as the positive or pump circulation system, is in most extensive use, and on large cars is almost universal. The other, known as the natural or thermo-siphon system, is used mainly on the smaller and cheaper cars.

In American passenger-car practice the use of the thermo-siphon system has been decreasing, for whereas in 1916, 70 out of 190 models then on the market had this system, in 1926 only 10 out of a total of 100 were equipped with it. In Europe, on the other hand, the thermo-siphon system is more widely used than the pump system. Besides dispensing with the pump, and, therefore, obviating the troubles caused by accidents to the latter, the thermo-siphon system has the advantages of varying the rate of circulation in proportion to the temperature of the cylinder wall and of maintaining circulation for a considerable time after the engine has been stopped, thereby reducing the liability of the radiator to freeze in cold weather.

Heat Absorbed by Jacket—According to numerous tests the proportion of the total heat in the fuel consumed absorbed by the jacket water at full load amounts on an average to about 32 per cent in the case of poppet valve engines—somewhat less in high-speed and somewhat more in low-speed engines. The brake thermal efficiency amounts to about 20 per cent, hence the heat absorbed by the jacket water is equal to about 160 per cent of the thermal equivalent of the full-load output of the engine. It is obvious that these figures can be only approximate, because both the thermal efficiency and the jacket losses vary with details of design. If the engine is of the valve-in-the-head type and has a nearly spherical combustion chamber, the jacket losses are somewhat lower than if it is of the L-head type, and especially if in the latter case the exhaust manifold is cast integral with the cylinder block and water cooled.

For a high-speed valve-in-head engine without internal exhaust manifold we may take the jacket loss at 28 per cent; for a high-speed L-head engine at 30 per cent and for a low-speed (not over 1200 ft. per min. piston speed) engine, 34 per cent. In a sleeve valve engine the jacket loss has been shown to be as low as 25 per cent. This is due to two causes—that the combustion chamber approaches the ideal spherical form, and that the cylindrical portion of the cylinder wall is separated from the flame by sleeves.

Calculating Jacket Heat—In calculating the amount of fuel consumed we may start with the maximum horse power developed by the engine and allow a fuel consumption of 0.67 lb. per hp.-hr. (which corresponds to a brake thermal efficiency of 20 per cent). Thus a six-cylinder 3 by 5 in. engine developing 60 hp. at 3000 r.p.m. would consume

$$60 \times 0.67 = 40.2 \text{ lbs. per hour.}$$

Assuming a heat value of 19,000 B.T.U. per pound of fuel, and that 30 per cent of the heat goes into the jacket, the amount of heat absorbed by the jacket water per minute is

$$\frac{40.2 \times 19{,}000 \times 30}{60 \times 100} = 3830 \text{ B.T.U.}$$

There are, of course, variable factors in the operation of every engine which affect the percentage of the heat going into the jacket water. Thus, for instance, late ignition and a rich mixture both cause increased heating. Such things, however, occur in the operation of every engine and must be allowed for in the factor of safety.

Size of Water Manifolds, Pump System—The inlet and return water pipes can be so proportioned that 3600 B.T.U.

are disposed of per minute per square inch of pipe cross-section—for pump circulation and passenger-car engines. In low-speed engines, where the pump action is apt to be less energetic, it is better to count on only 2400 B.T.U. per minute per square inch of section. On passenger cars water pipes of 1 in. inside diameter are used up to about 36 hp. and 1¼ in. with engines of larger output. On trucks the diameter is 1 in. up to 30 hp. 1¼ in. up to 55, and 1½ in. above 55 hp.

On the basis of the rates of heat transfer given, if we figure on a temperature drop of 20° in the radiator, 180 lbs. of water must flow through a section of 1 sq. in. per min. in the case of a passenger car, and 120 lbs. in the case of a truck. This is equivalent to water speeds of 415 and 275 ft. per min. for the two cases, respectively, which speeds would have to be attained at full engine speed.

Lengths of rubber hose are always inserted between the engine and the radiator at both top and bottom, so as to protect the radiator against vibration from the engine. The hose slips over fittings on the engine (or pump) and on the radiator tanks, for a distance of 1¼ to 1½ in. and a water-tight joint is effected by applying hose clamps. Hose and fittings have been standardized by the S. A. E.

Radiator Core Types—All radiators comprise a bottom tank and a top tank, between which is located the core, the real cooling or heat-dissipating element. In addition there is either a shell surrounding the tanks and core or a frame composed of the two tanks together with two side members or spacers.

Disregarding a type of core consisting of curved plain copper tubes extending between top and bottom tanks, which has been used to a certain extent abroad, all radiator cores may be divided into four classes, as follows: (1) Individual fin and tube; (2) continuous fin and tube; (3) ribbon cellular; (4) air tube cellular.

The first class mentioned, which is now used only on heavy trucks, consists of water tubes surrounded either by a crimped helical fin (Fig. 371) or plain washers, which are soldered to it. The tubes are assembled in header plates or sheets to which the tanks are soldered or bolted. With this type of core it is essential to have a rigid frame, resistant to lateral distortion, so as to obviate damage to the soldered joints.

Continuous fin and tube cores are now widely used. Referring to Fig. 372, horizontal sheets or fin plates with perforations are stripped over the vertical tubes, which latter are of either the seamless or lockseam type. The tubes are previously covered with solder and the fins are soldered to them by merely heating the assembled core in an oven to a temperature which will cause the solder to run.

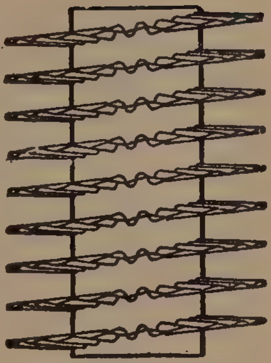


Fig. 371. — Finned Radiator Tube.

The tubes are of about ¼ in. diameter and the distance between them in the direction of depth is about the same, while transversely they are placed somewhat farther apart. In order to give a nicely finished appearance to the radiator, the front edge of the plates is folded over. This also stiffens the most exposed portion and lessens the danger of disfiguring the radiator. With continuous fin type radiators about 6 sq. in. of radiating surface is required per B.T.U. to be dissipated on passenger cars, and 10 on trucks.

In some designs of continuous fin and tube core, the tubes are directly in line in the fore-and-aft direction, while in other cases they are staggered. With the former arrangement the path for air flow is less obstructed, but with the latter the cool air is brought into more intimate contact with the metal. The tubes pass through header plates into which they are soldered, and in this case, too, a rigid frame is essential.

Ribbon cellular cores comprise really two types, those in which all of the sheet metal of the core is in contact with water on one, and air on the other side (all direct radiating surface), and those in which a portion of the sheet is in contact with air only (indirect radiating surface), receiving

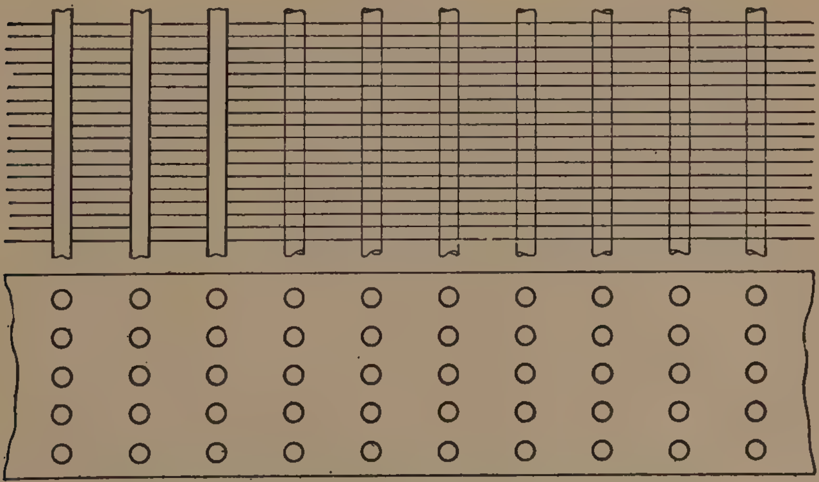


Fig. 372.—Continuous-Fin-and-Tube Assembly.

its heat through conduction from sheet in contact with the water. The water space is formed between two sheets with offset edges which are soldered together over a depth of 1/4-5/16 in., usually by dipping. Between the water channels spacers are arranged which generally serve to provide additional (indirect) radiation surface, while in other designs their chief object is to deflect the air currents into intimate contact with the water channel.

Radiators of cellular appearance are also made of flat tubing with seamed joints, the tubing being bent to form two or more sides of each vertical row of cells. Either all of the surface is direct cooling or else spacer sheets are used. With this core construction the path for the water from the top to the bottom tank is materially longer and there are fewer soldered joints. Various ribbon cellular core types are illustrated in Fig. 373.

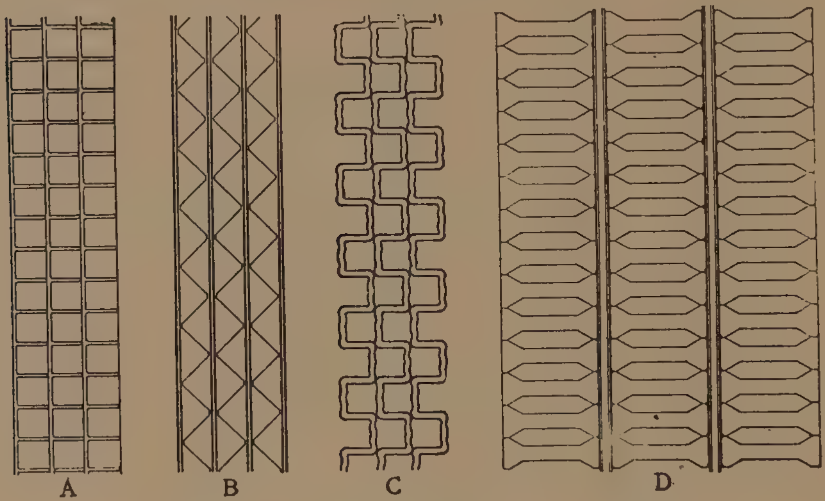


Fig. 373.—Ribbon-Cellular Type Cores.

With the air type cellular core, which consists (see Fig. 374) of a large number of short lengths of round, square or hexagonal tubes expanded at the ends and there soldered together by dipping, the water channels are formed by the spaces between adjacent tubes, while the tubes themselves form the air channels. All of the surface is in contact with water, but the water is more or less stagnant in the horizontal passages, which are therefore less efficient as cooling surface. At present the tubes are made with impressions in their walls to induce turbulent flow of both the air and the water, to add to the cooling capacity. The air tube cellular core is more expensive to manufacture than the other types, and therefore, is used mainly on high-priced cars.

Materials for Cores—The materials used for radiator cores include copper and two grades of brass containing 35 and 15 per cent of zinc and known as high and low brass respectively. Copper is used particularly for fins and spacers, on account of its high heat conductivity. It is very ductile and therefore can be readily worked in the press, but it has a lower tensile strength than the brasses. The high brass (containing a large proportion of zinc) is used to a considerable extent in ribbon radiators. It is not very ductile and must therefore be handled with care in the press. It has a tendency to be porous and care must be used not to overheat it in soldering.

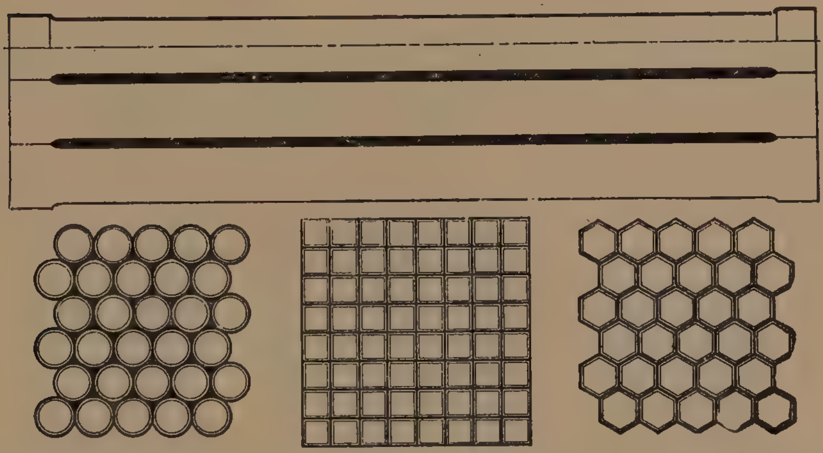


Fig. 374.—Construction of Air-Tube Cellular Type Core.

Low brass is considered the best core material by many manufacturers. It combines in a fair degree the ductility of copper with the rigidity of zinc, and it resists well the action of the alkalis found in the water in some sections of the U. S. The joints are made with either the standard tin solder (50 parts lead to 50 parts tin) or with a somewhat cheaper solder containing only 40 parts tin to 60 parts lead. This latter does not flow as freely as the standard solder.

The stock from which ribbon cores are made up varies in the thickness from 0.005 to 0.006 in., except in cases where the whole core is dipped in solder, when it is still thinner. The tanks are generally made of No. 18 to No. 20 B. & S. gauge stock.

Proportions of Cores—In ribbon and crimped tube cores the width of the water channel varies from 1/32 to 3/32 in., the average being about 1/16 in. Narrowing the water channel brings the water closer to the metal, which promotes heat abstraction, but in certain sections of the country very narrow channels easily become obstructed with lime. Moreover, the narrower the channel the greater the resistance to water flow. The water spaces are of such proportions that the average water content in 12 lbs. per cu. ft. of core. In connection with the width of water spaces it must be remembered that the water must descend through the core by gravity, and unless it descends as rapidly as water is drawn from the bottom tank by the pump at maximum engine speed, the top tank will fill up and water will be lost through the overflow.

Square tube cellular radiators are usually made of ¼ in tubing, and these have a theoretical radiating surface of 16 sq. in. per cu. in. of core, because a ¼ in. square tube 16 in. long has a volume of 1 cu. in. and an outside surface of 16 sq. in. However, the radiating surface is on the inside and the inside width of the tube is reduced by the width of the water space and the thickness of the stock. The actual radiation surface of air tube and ribbon cellular cores is from 10 to 12 sq. in. per cu. in.

The effective air passage area varies from about 55 to 85 per cent of the total frontal area, the average being about 72 per cent.

Heat Dissipating Capacity of Cores—The rate at which heat is dissipated by the radiating surface depends upon:

1—the rate of water circulation;
2—the air speed past the radiating surface;
3—the depth of the core;
4—the proportion of indirect to direct radiating surface.

At very low rates of water circulation the rate of heat dissipation is practically proportional to it, as the determining factor is then the rate at which heat is supplied. But with increasing speed of circulation the rate of heat dissipation increases less rapidly until at a water speed of about 20 ft. per min. through the core the rate of heat dissipation approaches a maximum. The pump and various other parts of the circuit should be so proportioned that this speed will be attained at a speed below full engine speed.

In the following, radiation surface refers to the total exposed surface, only one face of direct radiating surface and both faces of indirect surfaces being counted. Let the air speed through the radiator be S ft. per min. and let the core depth be d in. It is obvious that as the core depth increases the radiating surface becomes somewhat less efficient, for in passing through the air channels the air heats up and approaches the temperature of the metal surface, and its ability to abstract heat from the metal surface is thereby decreased. The heat dissipated per minute per square foot of radiating surface as above defined is given fairly accurately (for values within the usual limits of air speed and core depth) by the following equation:

$$H = 0.17S^{0.67} - 15.2(1/S^{0.27})d \text{ B.T.U.} \ldots$$

The effect of increase in radiator depth on the efficiency of the radiating surface is brought out in the following table taken from an S. A. E. paper by Karl F. Walker:

Table XIII—Increase in Heat Dissipation With Core Depth.

Increase in Depth of Core, in.		Air Velocity Through Radiator Core	
From	To	1800 ft. per min., per cent	4000 ft. per min., per cent
2	2½	19.2	21.8
2½	3	14.1	16.2
3	3½	11.0	14.0
3½	4	8.3	11.8
4	4½	6.3	9.5
4½	5	4.8	8.0

For a rough calculation of the depth of core required with air tube and ribbon cellular radiators it can be figured that 4 sq. in. of radiation surface are needed per B.T.U. per minute on passenger cars and 8 sq. in. on trucks, the need for a greater proportional surface on trucks being due to the lower air speed through truck radiators: Whereas on passenger cars the average air speed through the radiator at full engine speed will be of the order of 3600 ft. per in.; on trucks it will be only about one-half this.

Radiator Types—There are essentially three types of radiator, as follows: (1) The pressed shell type with removable core and tank assembly; (2) the cast tank type of radiator with either a removable or sectional core; (3) the sheet metal, integrally-built radiator.

The first class mentioned, with pressed metal shell, is almost the only one used on American passenger cars. Up to recently the shells were drawn exclusively from sheet steel and nickel-plated, but one manufacturer now uses duralumin sheet and another nickel, both of which eliminate the necessity for nickel-plating. Nickel-plated sheet brass is also used.

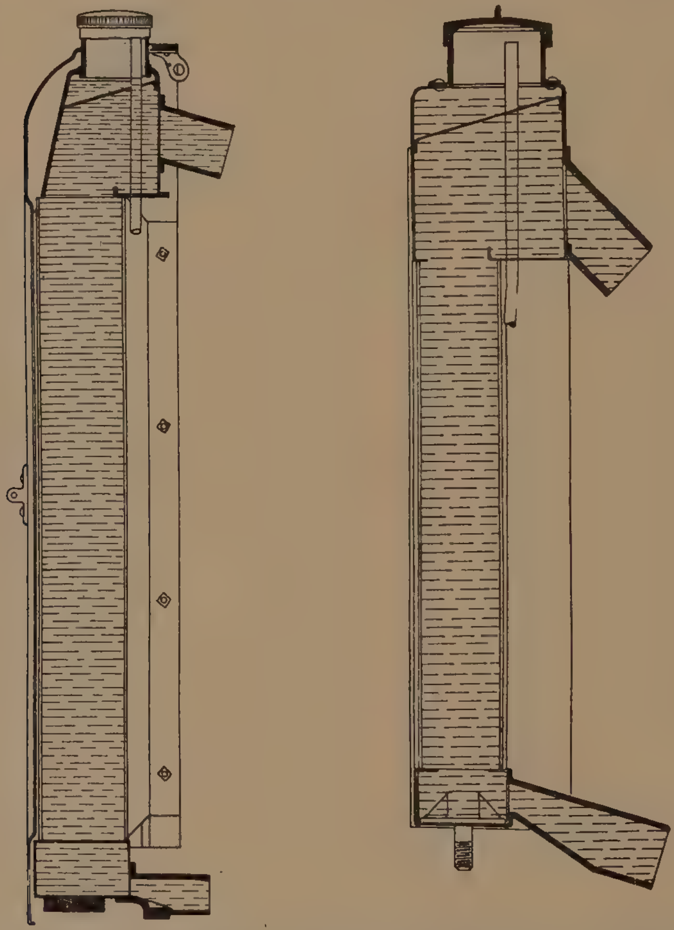


FIG. 375.—SECTIONS THROUGH RADIATOR WITH DETACHABLE PRESSED STEEL SHELL (LEFT) AND OF THE INTEGRALLY-BUILT SHEET METAL TYPE (RIGHT).

The depth of the shell is usually about 3 in. with an additional 1 in. for the hood ledge. For passenger-car radiators the shells are drawn from No. 18 to No. 22 material. Similar radiators are used for light trucks, in which case the shells are drawn from No. 16 to No. 20 stock. The pressed shell of the radiator permits of giving a neat form and finish to the front of the car, and it serves to support the core and tanks in such manner as to completely protect them against all strains due to distortion of the frame. Pressed steel side pieces are soldered to the side of radiator extensions running several inches over the tank ends. A cross-section through a passenger-car radiator with pressed shell is shown in Fig. 375 (on the left).

For mounting the complete radiator on the frame, a length of band iron is placed across the bottom of the shell so as to extend a short distance up its sides, and is riveted either directly to the shell or to brackets on the sides of the latter. The bottom of the shell either sets on a cross-member of the frame, to which it is secured by studs and nuts, or else the shell is provided with side brackets resting on the frame side members. If the lower tank rests on the supporting member, soft pads of woven material from 1/8 to 3/16 in. thick are interposed.

Cast Tank Radiators—In heavy truck work on solid rubber tires the radiators are subjected to great strains due to vibration, and radiators with cast tanks and frames are generally used. The cores are usually either of the finned tube or the continuous fin and tube type, and are assembled to header plates by soldering. The header plates are bolted up against the top and bottom tanks, between the side members and tanks at the sides, and with clamping strips behind the plate in front and rear. Gaskets are used to ensure water-tight joints. A sectional view of such a radiator is shown in Fig. 376 and a front view in Fig. 380.

With these radiators the most likely place for leaks to occur is at the joint between the tubes and header plates, and the tanks can then be removed and the leak repaired by soldering. Such radiators generally have a cushion or flexible support on the truck frame. The chief disadvantage of this type of radiator is its great weight. In high-speed work, as on buses, there is a tendency to reduce this by the use of cast aluminum tanks and frames, and with increasing production of trucks there is a possibility that pressed steel shells will be used for this class of work also.

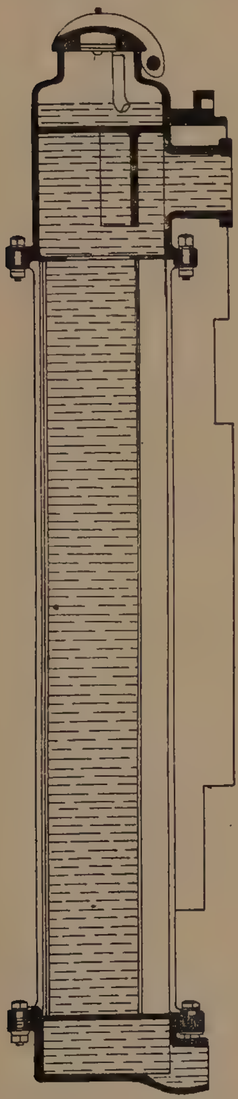
Sectional Core Radiators—In commercial vehicle work sectional core radiators have met with considerable favor, especially abroad. The core is made in six or more sections, each of which is bolted up against the top and bottom tanks in such a way that the water can flow through it from one tank to the other and that the joints are water tight. If any section becomes damaged it can readily be cut off from the two tanks so that the normal operation of the rest of the core is not affected. A section through a sectional core type of radiator is shown in Fig. 377.

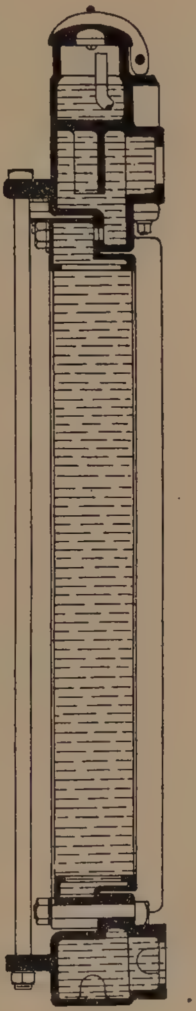
The integrally built sheet metal tank type of radiator (Fig. 375, right) is used rarely in automobile work, mainly where cars of a certain type are built in small numbers which do not warrant the expense of dies for a pressed steel shell.

Radiator Fittings—Water enters the top tank usually at the center and leaves the bottom tank near one side—the side on which the pump is located. The inlet and outlet fittings should be properly re-enforced. To ensure uniform flow of the water throughout the core, a baffle plate is sometimes provided in the top tank. An overflow pipe passes through the wall of the top tank at the rear and extends down the rear of the radiator near one side, its upper end abutting inside the filler tube. The filler tube is closed by a brass screw cap, which is now often molded in some composition having a low heat conductivity and having its gripping edge scalloped so as to enable one to get a firm hold. Many passenger cars carry either a radiator thermometer or an ornament on the radiator cap. On truck radiators hinged caps are sometimes used. A fine mesh wire strainer is placed in the filler tube to keep solid matter out of the system. To the back of the top tank is secured an anchorage, from which a brace rod runs either to the dash or to the engine.

Radiator Behind Engine—This arrangement originated with the Renault firm in France and has been used also by others, both in this country and abroad. The radiator takes the place of the dashboard and about one-half of it projects beyond the sides of the engine hood, which is of the characteristic Renault sloping type. The flywheel is provided with helical fan blades on its circumference, which exhaust the air from under the hood. Air can get into the engine space only by passing through that portion of the radiator outside the engine hood, into the cowl space, and thence through the remainder of the radiator into the engine space. From the engine space the air is exhausted by the flywheel fan. This system has some practical advantages, but it can hardly be said that it improves the appearance of a car. With no radiator in front many manipulations on the engine can be effected with greater freedom. The position of the radiator is favorable to thermo-siphon circulation, as the bottom tank is practically on the same level with the bottom of the cylinder jacket. Probably the greatest advantage over the front radiator is that it tends to keep the engine and accessories clean by drawing dust away from them instead of blowing it onto them.

The Thermo-Siphon System—In thermo-siphon cooling systems, the circulation is due to the fact that water is expanded by heat, and therefore a unit volume of hot water weighs less than a unit volume of cold water. A typical thermo-siphon system is illustrated in Fig. 381. When the engine is started up the heated water from the jacket begins to rise and its place in the jacket is taken by cold water from





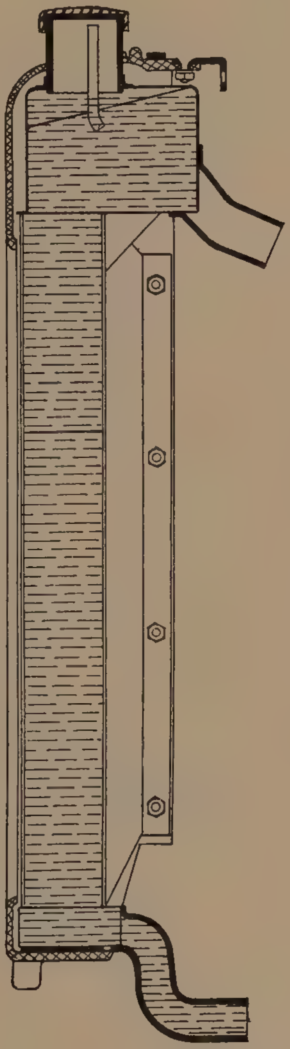


Fig. 376 (left).—Radiator with Cast Iron Tanks and Side Members. Fig. 377 (center).—Radiator with Sectional Core. Fig. 378 (right).—Radiator with Cast Aluminum Shell.

the bottom of the radiator. Circulation may be said to be due to the fact that the average temperature of the water in the column represented by the jacket and its connections is higher than the average temperature of the column represented by the radiator, and hence the average density is less in the jacket column than in the radiator column. The rate of circulation depends upon this difference in the average densities. The average density of the water in the jacket column is affected by the relative height of the jacket with respect to the radiator. The water return pipe is completely filled with hot water, while the pipe carrying the water from the radiator to the jackets is filled with cold water. Only about one-third of the height of the column is taken up by the jacket and the other two-thirds are taken up by the water pipes. Therefore, by placing the engine low relative to the radiator the head causing circulation can be increased. It is one of the conditions most essential to success with thermo-siphon cooling that the radiator be placed relatively high with respect to the cylinder jackets.

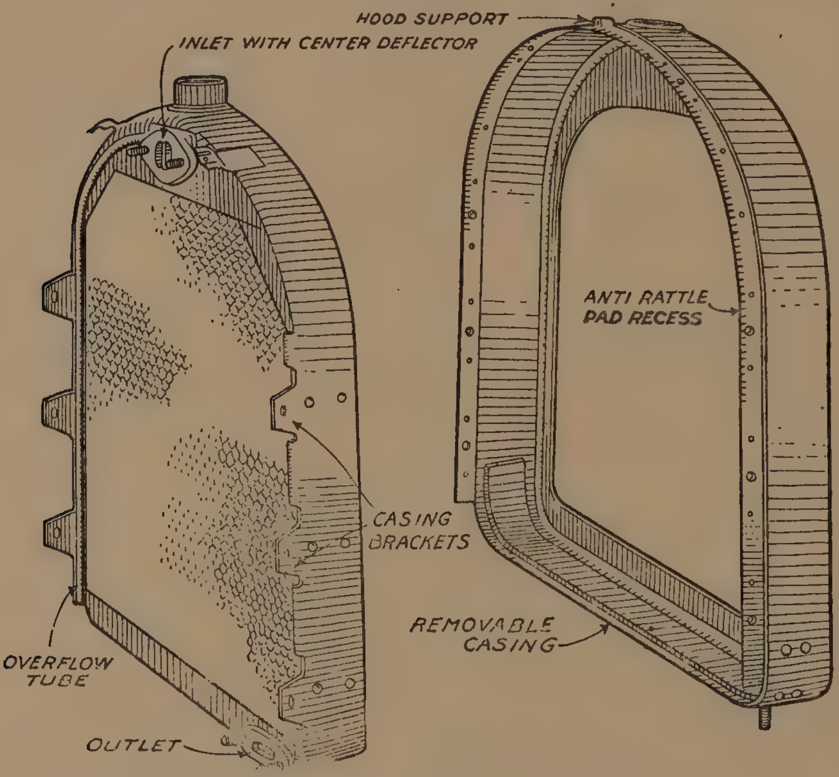


Fig. 379.—Pressed Steel Shell and Detachable Core.

Size of Pipes for Thermo-Siphon System—Even under the most favorable circumstances the circulation producing pressures are exceedingly small (of the order of one-hundredth of a pound per square inch cross-section of the column) and it is therefore necessary to make the resistance of the cooling circuit as low as possible. Tubular type radiators are usually employed with thermo-siphon systems. The pipes must not only be very liberal in size but the whole circuit must be made as short and as free from bends as possible. The system, therefore, is not so well adapted to six-cylinder and eight-in-line engines, and that probably explains why it is less popular in this country than abroad, where four-cylinder engines are in the lead. In determining the section of the pipes it is well to figure on a heat transfer of 600 B.T.U. per minute per square inch of cross-section at full engine load.

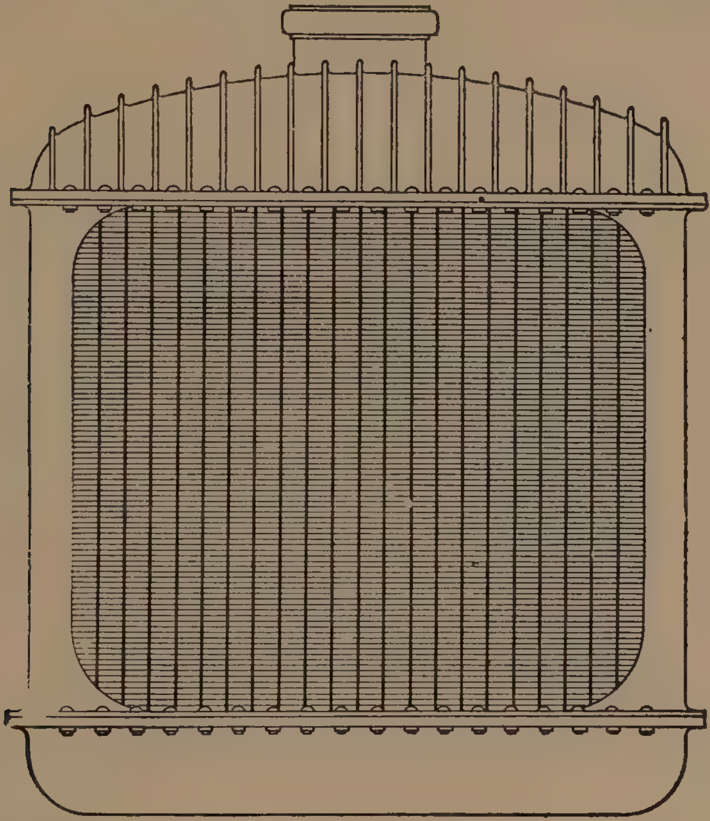


Fig. 380.—Front View of Truck-Type Radiator.

The top tank of a radiator for thermo-siphon circulation must have considerable water capacity, for the water will, of course, gradually evaporate and if the water level in the tank falls below the inlet from the jacket, then circulation is greatly hampered because the water from the jacket must be virtually raised over a wall. To secure this large water capacity it is customary to extend the top tank backward beyond the width of the radiator proper.

With the thermo-siphon system, owing to the much slower rate of circulation, the core radiation surface must be about 25 per cent greater than with the pump system. The temperature difference between the inlet and the outlet of the jacket will sometimes rise as high as 80° F.

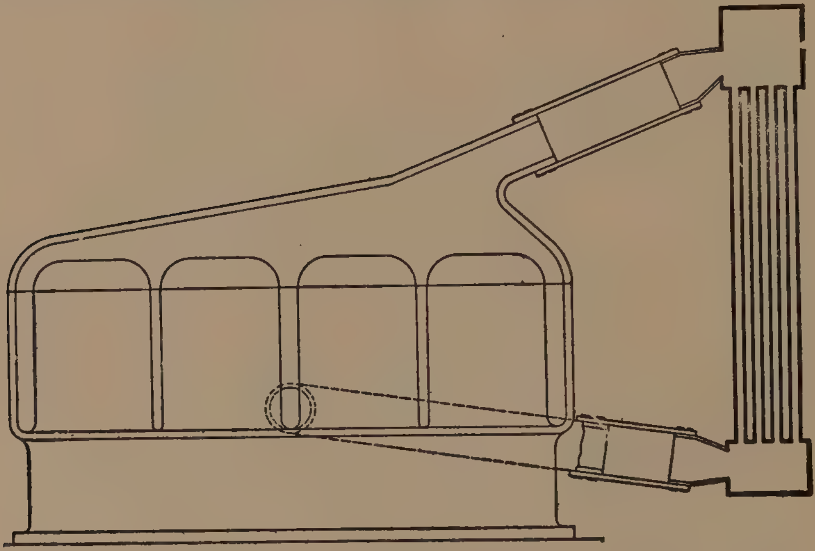


Fig. 381.—Diagram of Thermo-siphon Cooling System.

Fans—If a radiator had to dissipate all the surplus heat of the engine by radiation—as that term is ordinarily understood—it would have to be of enormous size. The process by which the transfer of the heat to the atmosphere is mainly effected is that of convection. That is to say, a large volume of air is drawn through the radiator, which, in coming in contact with the radiator surface, takes up heat that has been conducted through the cooling walls, and then carries it away with it. At high car speeds a great deal of air naturally passes through any front-mounted radiator, but automobile engines sometimes work under considerable load—and consequently pass much heat to the radiator—at comparatively low car speeds, and in order that the radiator may be able to dispose of the heat under these conditions, an engine-driven fan is mounted directly behind it.

Types of Fan—Radiator fans are made in two types, viz., fans with sheet steel blades riveted to a pressed steel or malleable iron center or hub, and cast aluminum fans in a single piece. The latter are the more recent type and are based on air propeller design.

Fans are made with from two to six blades, but the four-blade type is by far the most common. The earlier fans were generally made with flat blades, but, as will be shown later on, these are not as efficient as curved blades. With blades of considerable width near the tips there is a strong tendency to deform the blade at high speeds, and to counteract this it was formerly common to form the fans with an integral rim, the whole fan being pressed in a single piece, or to press corrugations in the blades. Both of these features tend to lower the efficiency of the fan, by setting up eddies in the air, and the more recent practice is to use smooth blades of a curved section as illustrated in Fig. 382, making them of slightly heavier stock.

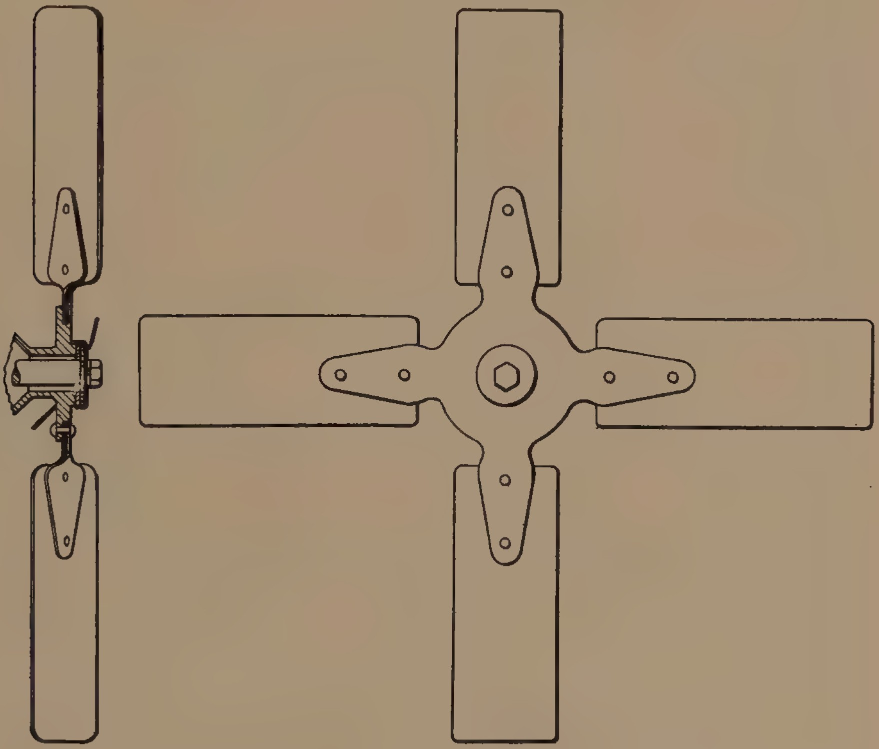


Fig. 382.—Four-Bladed Sheet-Metal Fan.

In cast aluminum fans the elements of the blades generally form helices around the axis. For instance, if we imagine a cylindrical surface of a diameter equal to the outer diameter of the fan, and co-axial with the latter, the outer edge of the blade will form a part of a helix on that surface. The distance along the cylindrical surface in a direction parallel with the axis in which the helix makes a complete revolution around its axis is known as its pitch. In a helical fan, as we pass from the outer edge of the blade inward toward the axis, the angle of the blade elements increases in the same measure as the distance from the axis decreases, so that the pitch remains the same. In an actual fan the blades, of course, are not mere surfaces, but have considerable thickness, and the curved surface above defined forms the central surface of the blades. Fig. 383 shows a two-blade helical fan and Fig. 384 the three-bladed one used on the Wills-Ste. Claire eight.

Theory of Fan Action—In order to develop equations for the air delivery and power consumption of fans we have to go to some extent into the theory of fan action. It is, of course, quite impossible to accurately calculate the air which will be drawn through a radiator on a given car, as this depends not only upon the relative "openness" of the radiator behind which the fan is mounted, but also very largely upon the freedom with which the air drawn in through the radiator may escape from the engine space. The fan blows the air directly against the engine, generator, carburetor, exhaust pipe, etc. At the bottom the engine space is enclosed by the mud pan, and the only outlet in many cases is between the flywheel housing and the pan. Wherever the outlets are located, they should have an aggregate area equal to the area of the air passage through the radiator, and preferably somewhat greater, as in passing through the radiator the air becomes heated and expands. If the outlet area is inadequate, a pressure builds up under the hood, which reduces the effect of the fan.

What the formulae which we shall develop will show is the air delivered by the fan, and the power represented by the air movement (the fan output) under ideal conditions. On a car, owing to the restricted air admission to the engine space and the more or less obstructed outlet, there will be a certain "slippage" or loss of air capacity, which in well worked out designs should not, however, exceed 30 per cent. The equations will permit of directly comparing fans of different numbers of blades, diameter, width, form and inclination of blades, as to air delivery and power consumption.

Air Resistance—Consider a plane of a square feet area to be moving in a direction normal to itself at a speed of

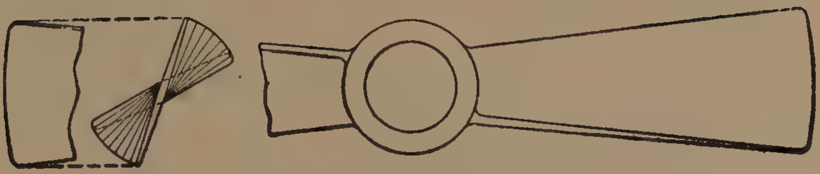


FIG. 383.—TWO-BLADED CAST ALUMINUM FAN.

v feet per second. Then in one second it will displace av cubic feet of air. This air will spill over the edges of the plane, as shown in Fig. 385, and at the moment will have a velocity

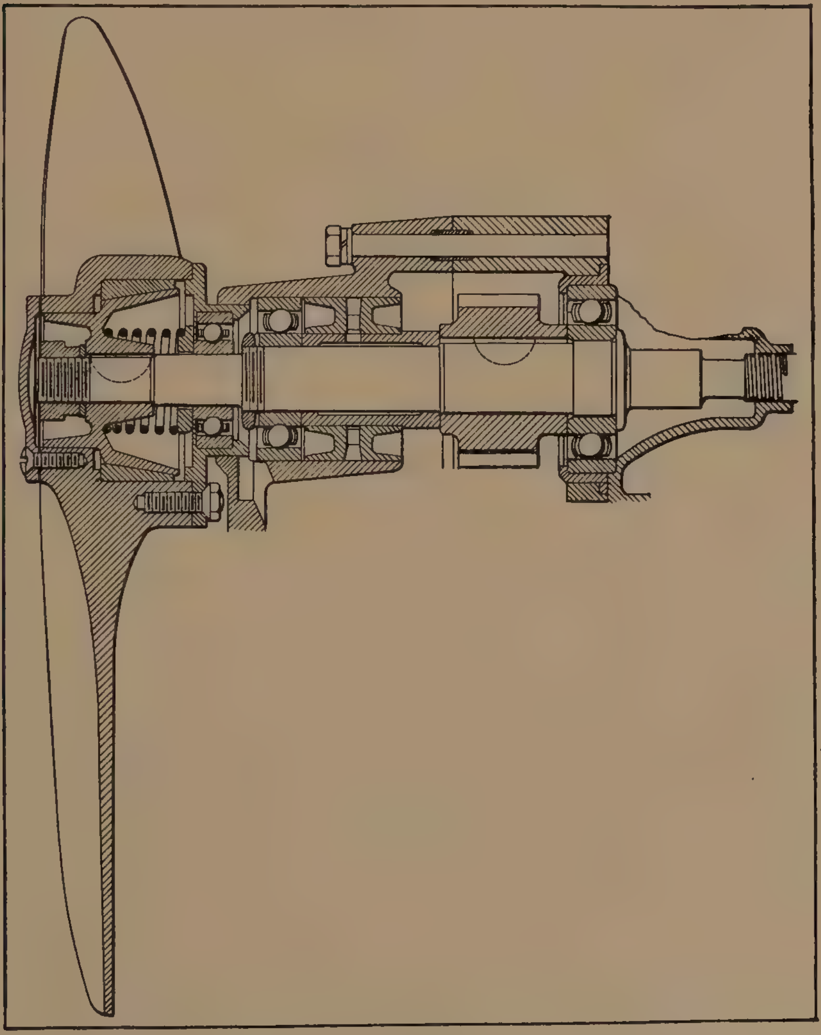


FIG. 384.—THREE-BLADED HELICAL CAST ALUMINUM FAN.

v in the direction of motion of the plane and the same velocity in a direction parallel to the plane. The resultant of these velocities is forward at an angle of 45° to the direction of plane motion, and the actual maximum velocity of the air is equal to the square root of the sum of the squares of its component velocities:

$$V = \sqrt{2}v$$

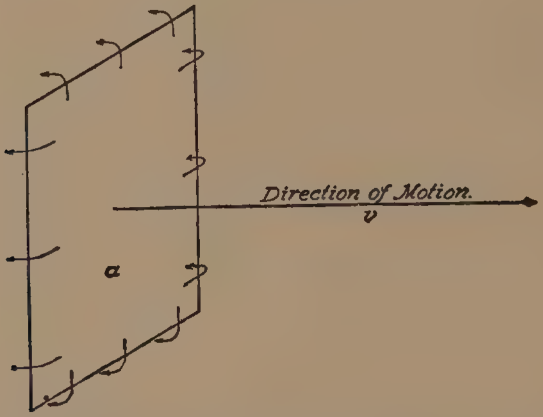


FIG. 385.—PLANE SURFACE MOVING STRAIGHT AGAINST THE AIR.

By reason of this velocity, the air has stored up a certain amount of kinetic energy, $mV^2/2$. Designating by W the weight of one cubic foot of air, the mass of air displaced per second is

$$m = \frac{Wav}{g}.$$

Therefore, the kinetic energy stored up

$$E = \frac{Wav}{2g}(\sqrt{2}v)^2 = \frac{Wav^2}{g}.$$

This energy, which is imparted to the air by the moving surface, is made up of two factors—the distance v moved by the surface and the resistance R encountered by it. Hence

$$R = \frac{\frac{Wav^3}{g}}{v} = \frac{Wav^2}{g} \text{ lbs.}$$

The weight of one cubic foot of air under normal atmospheric conditions is 0.077 lb., and the acceleration of gravity is 32.16. Hence

$$\frac{W}{g} = \frac{1}{418}$$

and

$$R = \frac{av^2}{418} \text{ lbs.}$$

Normal Resistance to Obliquely Moving Plane—Now let us take a plane surface (Fig. 386) moving in an oblique direction, the angle between the surface and the direction of motion being denoted by θ. We will use the same designations for the area and speed of the surface as in the previous case. Then, since the surface a is inclined at an angle θ, it receives only as much air as a perpendicular surface $a \sin \theta$ would. In other words, the speed of impact is only $v \sin \theta$, and substituting this for the speed in the equation for the pressure we get for the pressure on an inclined plane in the direction of motion

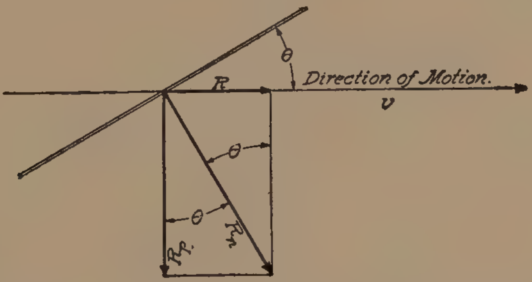


FIG. 386.—PLANE SURFACE MOVING OBLIQUELY.

$$R = \frac{av^2 \sin^2 \theta}{418}.$$

However, the reaction between the plane and the air is in a direction perpendicular to the plane, and the reaction in the plane of motion is only a component of the full reaction. Referring to Fig. 386, it will readily be seen that since

$$R_n = \frac{R}{\sin \theta},$$

the normal reaction is

$$R_n = \frac{\dfrac{av^2 \sin^2 \theta}{418}}{\sin \theta} = \frac{av^2 \sin \theta}{418}.$$

This expression is of great use in the investigation of fans. It is known as the law of von Loessl, and is often quoted in connection with problems relating to the sustaining force of airplanes. The pressure at right angles to the direction of motion which forms the other component of the normal pressure may be seen from Fig. 386 to be

$$R_p = \frac{av^2 \sin\theta \cos\theta}{418}.$$

We found above that when a surface moves straight on against the air the pressure developed is

$$R = \frac{av^2}{418}.$$

Here v may be regarded as the speed of impact between the stationary air and the moving plane. Now, in a direction at right angles to the direction of motion the plane is stationary and the air moves. Let the speed of impact in this direction, i.e., the speed of the air, be denoted by V. Then

$$\frac{aV^2}{418} = \frac{av^2 \sin\theta \cos\theta}{418}$$

and the speed of the air in the direction at right angles to the direction of motion of the plane is

$$V = v\sqrt{\sin\theta \cos\theta}.$$

We may now apply these principles to the case of a fan. In the ordinary radiator fan we have a plane blade making a certain angle with the plane of rotation. Referring to Fig. 387, we will take an element of this blade of radius x, the width of this element being dx, and the length w (the width of the blade). If we designate the speed of the fan in revolutions per minute by n, then the speed of this element is evidently

$$v = 2\pi xn \text{ ft. per min.}$$

x being given in feet.

Therefore, the speed of the air parallel to the fan axis will be

$$2\pi xn\sqrt{\sin\theta \cos\theta} \text{ ft. per min.}$$

and the element wdx of the fan blade will deliver

$$dV = 2w\pi n\sqrt{\sin\theta\cos\theta}\,xdx \text{ cu. ft. per min.}$$

Now, suppose that the blades are of constant width from the outer end, whose distance from the axis of rotation is R, to the inner end, whose distance from the axis of rotation is r. We then get the volume of air moved by the whole blade by integrating between the limits R and r, which gives

$$V = w\pi n(R^2 - r^2)\sqrt{\sin\theta\cos\theta} \text{ cu. ft. per min.}$$

If the fan has N blades, then the total volume of air moved per minute is

$$V = Nw\pi n(R^2 - r^2)\sqrt{\sin\theta\cos\theta} \text{ cu. ft. per min.}$$

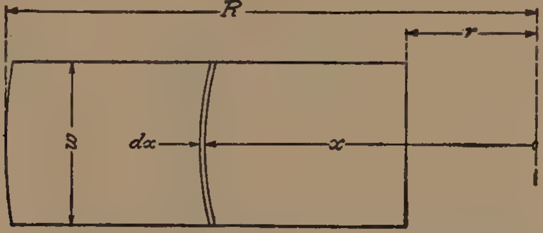


FIG. 387.—DIAGRAM OF FAN BLADE.

This is the theoretical delivery. It will be observed that with such a fan the volume of air moved per minute varies directly as the number of blades, the speed, the width of blade and a function of the blade angle, and nearly as the square of the fan diameter, since the inner radius is usually very small as compared with the outer radius.

Owing to the very irregular nature of the passage through which the air is forced on an automobile it is very difficult to calculate the volumetric efficiency or the proportion of actual to theoretical air delivery of a fan. Rather extensive tests of fans of different designs have shown that the volumetric efficiency is about 70 per cent when the minimum area of the outlet passage is equal to the area of the fan; 62 per cent when it is three-fourths that of the fan; 52 per cent when it is half that of the fan, and 35 per cent when it is one-fourth that of the fan.

If the width of the blades increases toward the tip, the actual width at any radius x is $a + bx$, where a and b are constants. It follows directly from a development of the foregoing principles that the delivery of such a fan is

$$V = 2N\pi n\sqrt{\sin\theta\cos\theta}\left[\frac{a}{2}(R^2 - r^2) + \frac{b}{3}(R^3 - r^3)\right] \text{ cu. ft. per min.}$$

To find the air delivery of a helical fan it is necessary to divide the length of the blade into a considerable number of equal sections, determine the mean angle for each section, calculate the air delivery for that section by assuming it to have a constant angle equal to the actual mean angle, and then adding the deliveries due to the different sections.

Power Absorbed by Fans—We found above that the component of the normal pressure on an obliquely moving plane in the direction of motion is $av^2 \sin^2 \theta/418$. Therefore, the component of the normal pressure in the direction of motion on an element wdx of a fan blade is

$$\frac{v^2wdx \sin^2 \theta}{418} \text{ lbs.}$$

The velocity of the element is

$$v = \frac{\pi xn}{30} \text{ ft. per sec.}$$

Hence

$$v^2 = \frac{\pi^2x^2n^2}{900}.$$

Substituting this value of v^2 in the above expression for the component of the pressure in the direction of motion, we get

$$dF = \frac{\pi^2x^2n^2}{900} \times \frac{wdx \sin^2 \theta}{418} = \frac{\pi^2n^2w \sin^2 \theta}{376{,}200}x^2dx.$$

The moment of this force around the axis of rotation is found by multiplying both sides of the equation by x.

$$xdF = dM = \frac{\pi^2n^2w \sin^2 \theta}{376{,}200}x^3dx.$$

Integrating between limits R and r we get the total moment due to one blade:

$$M_1 = \frac{\pi^2n^2w \sin^2 \theta}{1{,}504{,}800}(R^4 - r^4).$$

If there are N blades we must multiply by this figure to get the total moment for the whole fan, and the horse power equivalent of the output is then found by multiplying by $2\pi n$, and dividing by 33,000 –

$$\text{H.P.} = \frac{N\pi^2 n^3 w \sin^2\theta}{25{,}000{,}000{,}000}(R^4 - r^4).$$

The input in horse power is, of course, somewhat greater, because there is frictional loss at the bearings and, besides, the fan blades do not consist of mere surfaces, but of plates of some thickness whose forward edges stir up the air and thereby create a loss.

The above equation also assumes that the fan is working in free air. Restrictions of the inlet and the outlet have opposite effects on the power absorbed, and may therefore neutralize each other. Restriction of the inlet causes the fan to revolve in a partial vacuum, and reduces the power consumption, while restriction of the outlet causes an over-pressure in the space around the fan, and increases the power consumption.

In a fan with plane blades of tapering width in which

$$w = a + bx$$

the torque moment due to an element of the fan blade is

$$dM = \frac{\pi^2 n^2 \sin^2\theta}{376{,}200}(ax^3dx + bx^4dx)$$

and the horse power consumed is

$$\text{H.P.} = \frac{N\pi^3 n^3 \sin^2\theta}{6{,}200{,}000{,}000}\left[\frac{a}{4}(R^4 - r^4) + \frac{b}{5}(R^5 - r^5)\right]$$

For a helical fan the power absorbed is given by the following equation

$$\text{H.P.} = \frac{Nn^3\pi wp^2}{49{,}660{,}000{,}000}$$

$$\{R^2 - r^2 - (p^2/4\pi^2)\log_n[(R^2 + p^2/4\pi^2)/(r^2 + p^2/4\pi^2)]\}$$

which must be divided by the efficiency factor to get the power input. It appears from some fan tests of which the author has the data that an efficiency of 75 per cent can be figured on. In the above equation, as in all others in this section, all linear dimensions must be in feet. The natural logarithm, which occurs in the equation, is found by multiplying the ordinary logarithm by 2.303.

Concave Blades—It was discovered by Lilienthal, one of the pioneer experimenters in the field of flight, that the air pressure on a surface which is inclined to the plane of motion is greatly increased by making that surface concave on the side which repulses the air. The gain is especially great if the angle of the surface with the plane of motion is small. The surfaces investigated by Lilienthal had a section in the form of a circular arc, and are defined by the ratio of the height *h* to the length *s* of the chord. Lilienthal investigated surfaces of three degrees of concavity—1:12, 1:25 and 1:40. In the case of the concaved surfaces the angle is that between the chord of this surface and the plane of rotation. It is a rather peculiar fact that with the degree of concavity 1:12 there is no appreciable variation in the air pressure, and hence in the air delivery, for angles of 20° to 45°, and with surfaces of smaller concavity the variation in the air delivery, if the angle is changed from one of these extremes to the other, is also much less than in the case of plane surfaces. In the experiments which led to the results recorded in Fig. 388 the surfaces were moved in a circular path of 23 ft. diameter, at speeds up to 2400 ft. per min. All of the surfaces used had the same form, and an equal area of 5.38 sq. ft. Lilienthal's experiments proved further that for degrees of concavity greater than 1:12 the air delivery decreases again. It also appears from his experiments that at higher speeds than 2400 ft. per min. the best concavity is less than

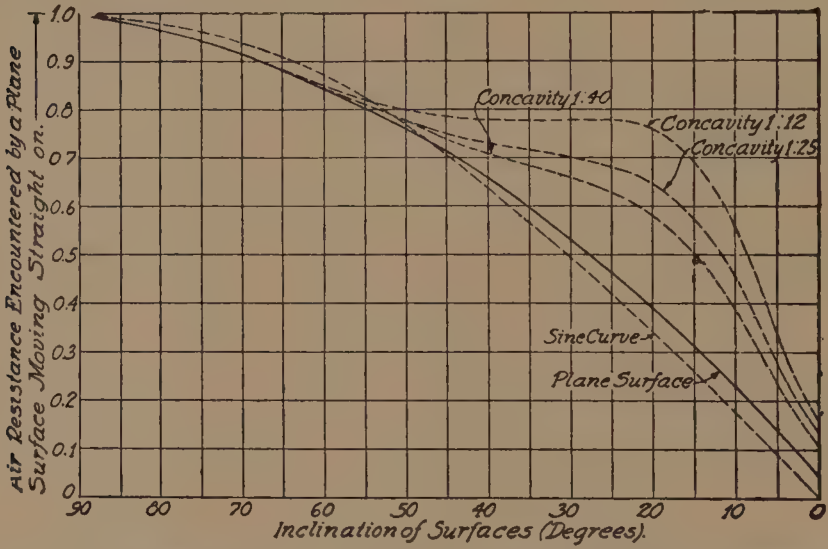


Fig. 388.—Pressure of Concave Surfaces Moving Obliquely through the Air (Lilienthal's Experiments).

1:12, and that for each speed there is a corresponding optimum concavity.

The tests made and the results recorded bear on the full pressure against the surface. Except for very small angles of inclination, the direction of this pressure is substantially perpendicular to the chord. If this pressure is multiplied in a certain ratio its components in the plane of motion and at right angles to this plane are multiplied in the same ratio. This means that the power consumed increases in proportion to the normal pressure, and the thrust at right angles to the plane of motion the same. The latter is proportional to the square of the air delivery. Hence the air delivery will be increased by concaving the blade, in a ratio which is equal to the square root of the ratio in which the normal pressure is increased. Curves showing the proportional increase in air delivery, due to various degrees of concaving with different blade angles, which were derived from the Lilienthal curves, are shown in Fig. 389. Concaving the blades, however, has another advantage besides making it possible to obtain a greater air delivery from a fan of given blade dimensions, number of blades and speed of revolution. It insures a higher

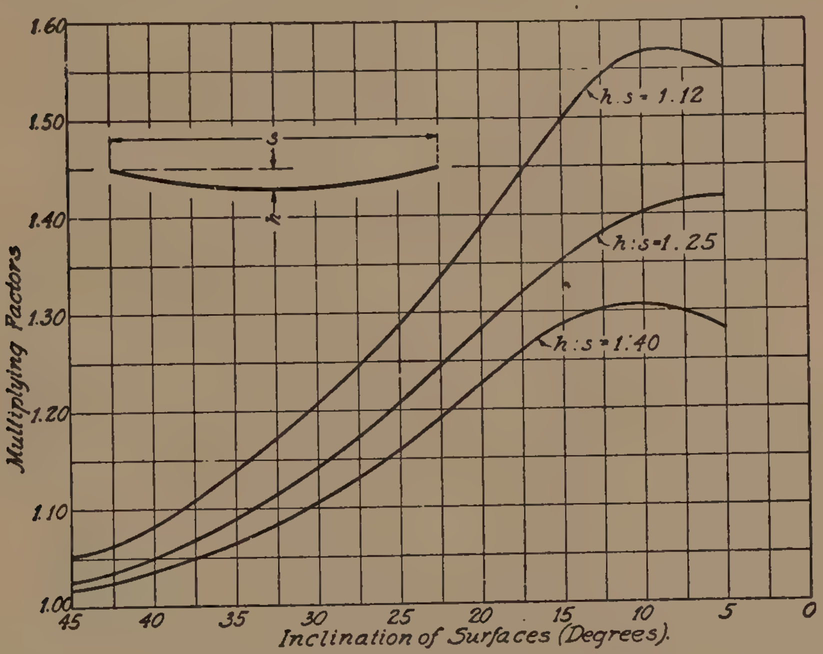


FIG. 389.—DIAGRAM SHOWING INCREASE OF AIR DELIVERY DUE TO CONCAVING FAN BLADES.

efficiency; that is, less power consumption for a given air delivery.

It will have been noticed that with each form of blade the air delivery is directly proportional to the speed. On the other hand, the power consumed increases as the cube of the speed. Thus, for instance, if we double the speed of the fan we get twice as much air per minute, but the power consumption will be eight times as great. Now, by concaving the blades of a fan we can increase the delivery of that fan in a certain ratio, which we will designate by r. Then the power consumption of that fan will also increase in the ratio r. However, if only the original air delivery is required, then, if we designate the speed at which the straight-bladed fan gives this delivery by v the concave-blade fan to give the same delivery needs to run only at a speed v/r. If we designate the power consumed by the straight-bladed fan by P, then that consumed by the concave-bladed fan at the same speed is Pr, and for the same air delivery as that of the straight-blade fan this fan will consume

$$Pr \times \frac{1}{r^3} = \frac{P}{r^2}$$

which is less than P, since r is greater than unity. This proves the greater efficiency of the concave blades.

Fan Capacity Required—Since at high speed the motion of the car induces sufficient air circulation to abstract from the radiator all the heat that enters it, it is obvious that the fan capacity per engine horse power or per B.T.U. to be disposed of need not be as great in a high-speed car as in a low-speed one. Data of the fan equipment of a number of tractors, trucks and passenger cars when averaged shows the following theoretical air capacities per B.T.U. to be disposed of in the radiator at full engine load for different classes of vehicle:

Table XIV—Dependence of Radiator Capacity on Car Speed

Maximum Vehicle Speed	Cu. Feet of Air Per B.T.U. Entering Radiator
5 m.p.h.	1.75
15 m.p.h.	1.5
30 m.p.h.	1.25
45 m.p.h.	1
60 m.p.h.	0.90
70 m.p.h.	0.80

If the engine is of the ordinary L-head type the B.T.U. to be disposed of may be quickly found by multiplying the maximum brake horse power by 68, and by multiplying the product thus obtained by the fan capacity factor corresponding to the maximum speed of the vehicle, taken from the above table, we have the necessary theoretical air capacity of the fan in cubic feet per minute at the speed corresponding to the engine speed of maximum output.

In determining the diameter of the fan it must be remembered that only that portion of the radiator covered by its sweep is benefited by it; and for this reason, and also because a large size fan is more efficient than a smaller one of the same air delivery, it is well to choose the largest diameter practicable. However, 20 in. seems to be the limit for road vehicles. On small passenger-cars fans of 13 and 13½ in. diameter are used. The 20-in. fans are used for the largest truck engines of about 50 hp. The air delivery, which should be in direct proportion to the horse power, varies as the cube of the linear dimensions of the fan (as D^2 and as w), hence if a 20-in. fan is suitable for a 50-hp. engine an 18-in. fan is suitable for a 36 hp. and a 16-in. fan for a 25.5-hp. engine of the same type. As between a fan of smaller diameter covering only a relatively small fraction of the radiator surface and a larger fan running at somewhat lower speed, the

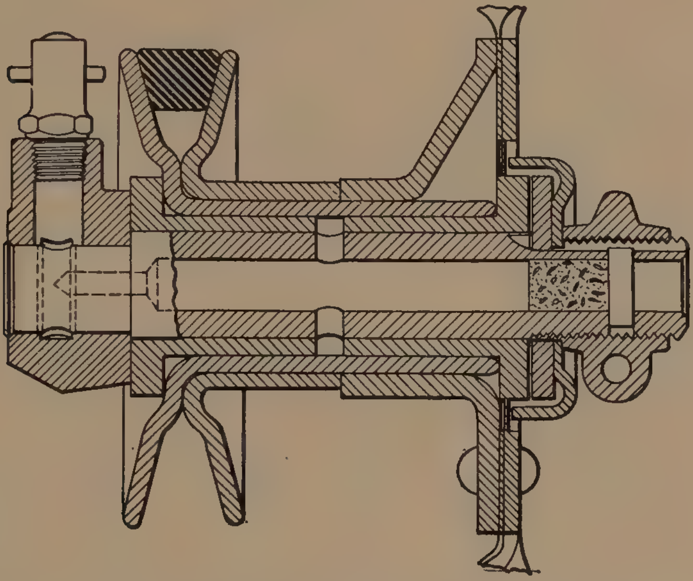


FIG. 390.—PLAIN-BEARING FAN HUB.

advantage lies undoubtedly with the latter. The highest efficiency is obtained if the air speed is the same over the whole area of the radiator.

Practically all of the different types of bearing are used in fans. For the lower priced cars it is customary to use plain bearings. A fan hub of this type with pulley for V belt is shown in Fig. 390. The spindle or stud is drilled out axially from the forward end, to serve as an oil reservoir. It is riveted into the fan bracket, into which is fitted an oiler having an outlet communicating with the oil chamber inside the stud by a circumferential groove in the hole in the bracket and a radial hole in the stud.

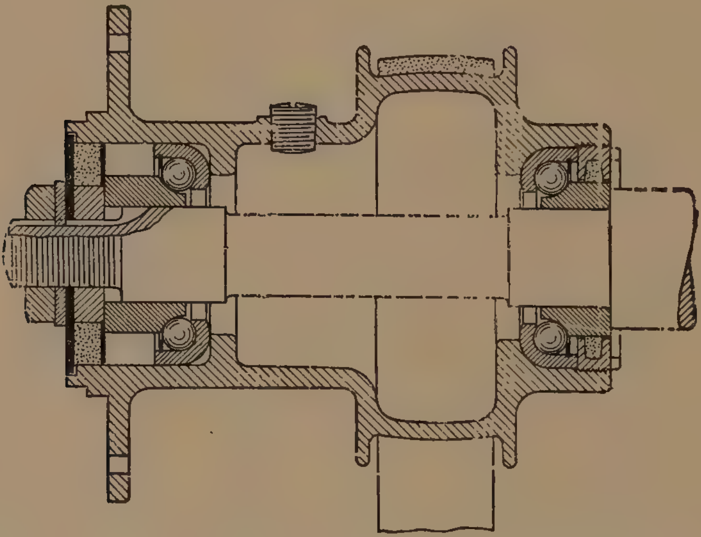


FIG. 391.—FAN HUB WITH CUP-AND-CONE TYPE BALL BEARINGS.

The hub and pulley together are made up of three flanged tubes or sleeves. The flanges of two of these sleeves, of which one telescopes over the other, form the pulley for the V belt. The inner of these two tubes has a flanged bearing bushing inserted into it from each end, while on its outside it carries, in addition to the second member of the belt pulley, a flanged sleeve to which the fan center is riveted.

On the hub of a radiator fan there is, of course, a certain amount of end thrust, the fan normally being drawn toward the radiator; but at high car speeds the thrust is likely to be reversed, the car speed being higher than the air speed due to the fan, and provision must be made for thrust in both directions. In the design shown the forward thrust is taken up on a large thrust washer which is keyed on the stud.

Adjustment of belt tension is by the nut on the right. In the drawing the two sides of the V belt pulley are shown close together, and this is the limit of adjustment. When a new belt is applied the two sides are farther apart and the belt rides near the bottom of the groove. Then, by screwing up the adjusting nut the two sides of the pulley can be brought close together and the belt thereby tightened. A pressed steel hub cap is inserted between the thrust washer and adjusting nut.

Figure 391 shows a fan hub with cup-and-cone type ball bearings. These were much used in earlier years but have lost considerably in popularity, for one reason because they are often overloaded by injudicious adjustment of the belt tension and are then easily damaged. Referring to the illustration, fan hub and pulley are cast integral, in accordance with common practice. The hub is formed with two internal flanges, against which the cups of the cup and cone bearings abut. The cone of the rear bearing is pressed over the shaft against a shoulder on the latter and the cone of the forward bearing is adjusted in position by means of a nut on the end of the threaded shaft, which is provided with suitable locking means. The forward end of the hub is always closed, either by a metal cap or by a felt washer and ring, so that none of the dust drawn in through the radiator can get into the bearings.

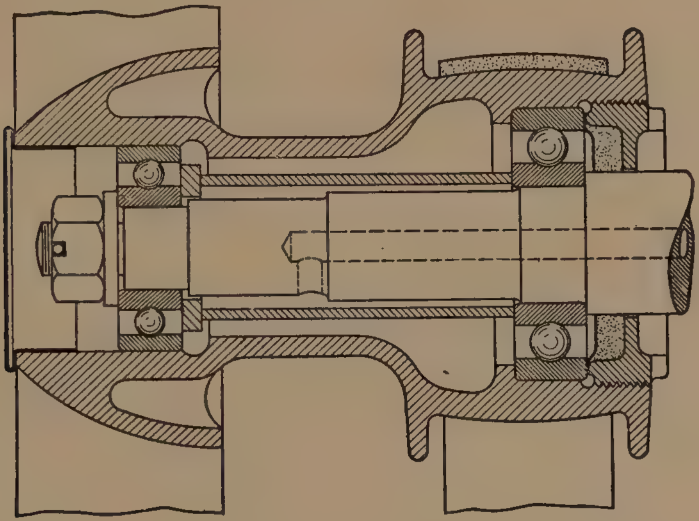


Fig. 392.—Fan Hub with Annular Type Ball Bearings.

A fan hub and pulley design employing annular ball bearings is shown in Fig. 392. In this an attempt is made to reduce the central dead area of the fan to a minimum by reducing the diameter of the hub toward the front. Both of the inner ball races are fixed on the shaft longitudinally by a spacer and clamp nut, but only the rear one of the outer races is fixed in position, and this bearing, therefore, takes all the thrust load. The pulley is flanged, as is common practice. This fan is of cast aluminum, and blades, hub and pulley are a single casting. In other designs of cast fan the fan proper is cast separately and bolted to the end of the combined hub and pulley.

Roller bearings are now extensively used in fan hubs in American practice, particularly the flexible roller type. A mounting embodying these bearings will be described farther on when dealing with combined fans and pumps.

Fan Drives—On most cars the fan is driven by belt. There are generally three shafts, to any one of which the driving pulley may be secured—the crankshaft, the camshaft and the magneto or generator-driving shaft. An objection to placing the pulley on the crankshaft is what might be called a traditional one. In power transmission on a large scale belts are never arranged vertically, because in a vertical drive the weight of the belt tends to reduce the friction between it and the lower pulley. The weight of the belts often used in power plants is such that it alone will give sufficient grip on the pulley and the belt may be run comparatively loose. However, the weight of a fan belt is insignificant, as compared with the pressure with which the belt must be applied to the pulley, and the objection mentioned therefore is void in this case. It is rather seldom that the driving pulley is secured to the camshaft, because the fan generally runs at from two to three times the speed of the camshaft and a large driving pulley would be required on this shaft, as the diameter of the driven pulley is determined by the power to be transmitted. The preferred location of the fan-driving pulley is on the magneto or generator driving shaft, which turns at crankshaft speed in four cylinder and one and one-half times crankshaft speed in six-cylinder engines, and therefore requires only a small and light pulley.

Belt Drive Practice—Flat, round and V-type or trapezoidal belts are used for fan drives. Flat belts are used particularly on trucks, and the four-blade 20-in. fans used on the larger truck engines usually take a 2-in. belt, while small passenger-car and light truck engines take a 1-in. belt. Belt speeds vary from about 1500 ft. per min. on passenger cars with low-speed engines to about 4500 ft. per min. on passenger cars with high-speed engines, intermediate speeds of from 2000 to 2500 ft. per min. being used on the larger trucks.

Pulleys for both flat and V belts have been standardized by the S. A. E. The minimum diameter of the driven pulley is 3 in., the minimum diameter of the driving pulley 3 in. for a drive ratio of 1½ to 1 (geared up); 5 in. for 1 to 1 ratio, and 7½ in. for a ½ to 1 ratio. Pulleys for flat belts are made with a crown of a minimum radius of 7 in. and flanges of 3/16 in. minimum width and height (these latter in the case of cast pulleys only). At the junction of the flange and face a fillet is provided, with a minimum radius of 3/32 in.

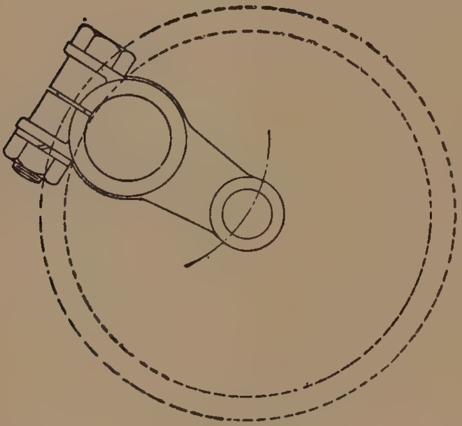
Two angles have been standardized for pulleys—38° when a fan alone is to be driven, and 28° when some other accessory is to be driven in addition. The width of the groove at the outside diameter of the pulley must be greater than the maximum width of the belt. For link belts (which are now rare) the next larger size pulley is used. The included angle of the sides of rubber V belts is 32° when another accessory is to be driven in addition to the fan, and 42° when intended for fan drive only. This is 4° more than the angle of the corresponding pulley, and allows for the bulging of the inside of the belt when laid around the pulley. The rubber type of V belt has come into extensive use recently and is giving very satisfactory service. In fact, since the introduction of this type of belt several manufacturers have adopted belt drive for the generator, which is sometimes placed in tandem with the fan.

Whenever belt drive is used provision must be made to take up any slack in the belt. This is generally accomplished by means of a swinging supporting arm or an eccentric device. The former method is illustrated in Fig. 393 and the latter in Fig. 394. Where the fan stud is supported by a bracket secured to the cam gear housing the latter can be provided with vertical slots so that the stud can be shifted vertically.

On a number of engines the belt is tensioned automatically. The stud for the fan pulley is then secured to one arm of a lever which pivots on a stud screwed into the engine block or is supported by the gear housing cover, the other arm of the lever being subjected to adjustable spring tension. The arm to which the spring is connected is usually considerably longer than that carrying the hub spindle, so the belt tension will be several times the spring tension.

Positive fan drives, which are used with V type and overhead camshaft engines, were referred to in Chapter XIII. An example of such a drive is the Wills, illustrated in Fig. 384. When a fan is driven through a positive transmission, such as a chain or gears, it must have a slipping clutch incorporated in its hub, which will slip and prevent excessive strains from high acceleration or deceleration of the engine. The clutch usually consists of two or three conical or disc members which are pressed together by a coiled spring and transmit the driving torque by virtue of their surface friction.

Fan Shroud—In the average car the fan sweeps over little more than one-half of the total area of the radiator, and at low car speeds those portions not covered by the fan sweep dissipate comparatively little heat. To increase the efficiency of the radiator, on motor trucks the fan is sometimes surrounded by a shroud, as illustrated in Fig. 395. The shroud is secured to the hood ledge and the fan must be placed a sufficient distance back of the core (2½ in. or more), so that air passing through the outer portions of the core will not be impeded too much by having to turn at a very sharp angle. For maximum efficiency the rear edges of the fan blade should be in a plane with the edge of the shroud.



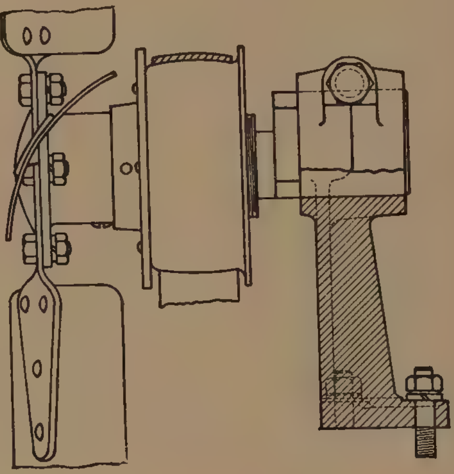


Fig. 393 (left).—Fan-Belt Adjustment by Means of Swinging Bracket.

Fig. 394 (right).—Fan-Belt Adjustment by Means of Eccentric Stud.

If adjustment must be provided for the opening in the shroud is made about 1½ in. larger in diameter than the fan, as illustrated in Fig. 395. Fans not provided with a shroud should not be placed closer to the radiator than ½ in., as they tend to be noisy when too close.

Pumps—Only one kind of pump is used for circulating the cooling water, the centrifugal type, which is generally the best when comparatively large quantities must be moved against a low pressure head. The maximum rate at which the water must be circulated in any particular car can be easily calculated from the amount of heat to be dissipated and from the permissible temperature drop in the radiator. This latter is usually assumed to be 20° F., though some authorities place it as low as 15° F. The number of B.T.U.s to be dissipated per minute is equal to 68 times the number of brake horse power, under favorable conditions, but radiator and fan engineers often make an allowance of about 20 per cent for unfavorable conditions, such as rich mixture, late spark, slipping fan belt, etc.

Granting that the favorable conditions obtain normally, if the temperature drop in the radiator is limited to 20°, the rate of circulation evidently must be 68/20 = 3.4 lbs. or 0.42 gal. per horse power per minute.

If the rate of circulation were very low, the temperature drop in the radiator would be greater. This would reduce the mean temperature of the water in the radiator, and, consequently, the difference between this mean and the mean temperature of the air on passing through the radiator, to which difference the dissipation of heat by the radiator is directly proportional. Rapid circulation also permits of a higher mean jacket temperature, which tends toward a higher fuel economy.

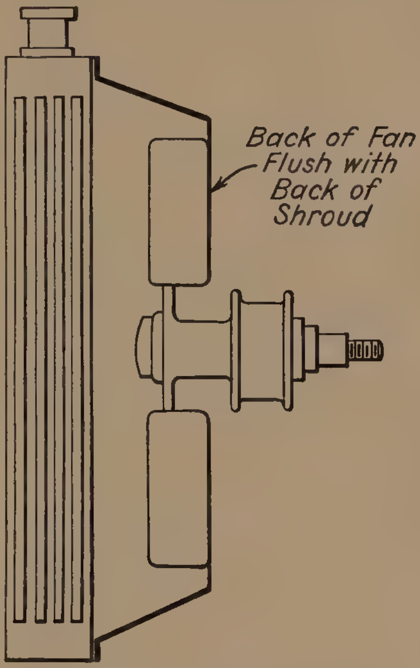


Fig. 395.—Fan Shroud.

Since the water descends through the radiator core by gravity more quickly than it is moved by the pump, the pressure head against which the pump must move it depends mainly upon the distance from the outlet of the bottom tank to the inlet of the top tank.

Centrifugal Pumps—A centrifugal pump comprises two main parts, viz., the impeller and the housing. The impeller contains a set of straight or curved vanes which sometimes are supported only by the hub, but more frequently by a disc at one side. The housing, which must be made in two parts so that the impeller may be inserted into it, is provided with a water inlet concentric with the shaft on which the impeller is mounted, and with a water outlet at the circumference. In a cooling system on a car the water, of course, fills the whole pump chamber, the pump being always located below the level of the top tank. Rotation of the impeller sets the water between the impeller vanes in motion, and the centrifugal force resulting from this rotation causes the water to flow outward toward the circumference of the housing where it enters the volute—a water passage leading entirely around the circumference of the housing and gradually increasing in cross-section from almost nothing to the full area of the discharge opening.

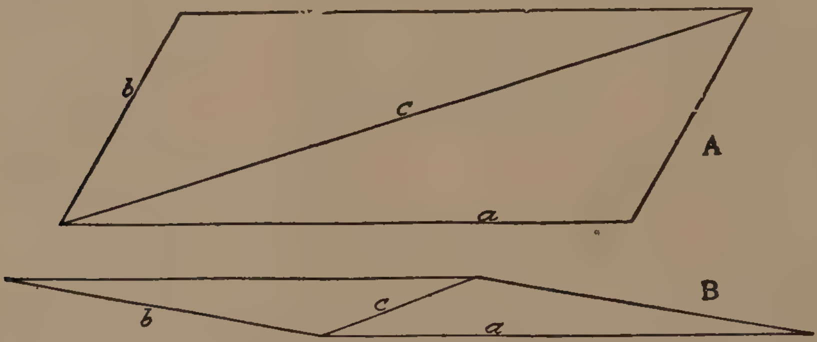


Fig. 396.—Diagrams of Water Flow in Centrifugal Pumps.

Angle of Vanes—Inside the pump the water has a dual motion; or, rather, its motion can be resolved into two components. It will rotate with the impeller and it will also flow parallel to the impeller blades from the central inlet toward the volute. The actual motion of the water, upon which the pressure impressed upon it by the impeller blades depends, is the resultant of its rotational motion with the impeller blade, and of its motion relative to the blade, parallel to the outer section of the latter, and by changing the direction of the vanes we can change the speed and direction of motion of the water for any given impeller speed. This is illustrated in Fig. 396.

The two parallelograms of motion correspond to impellers for high pressure (A) and low pressure (B) centrifugal pumps. In each diagram the vector *a* represents the speed of the impeller rim, the impeller tending to carry the water with it. In each also, *b* represents the speed of the water parallel to the vanes. The resultant of the two components is *c*, the actual speed of the water. It will be seen that in case B, where the vane is turned in the direction opposite to that of rotation, the actual speed of the water is very much less than in case A, where the vane is turned in the direction of motion.

The pressure head against which the pump is capable of moving the water is proportional to the square of the velocity of the water. On the other hand, the resistance to the flow of water through the pump is much less if the pressure produced is less, that is, when the vanes are curved backward and, therefore, if only a small head has to be worked against, a pump with backwardly curved vanes will be the most efficient.

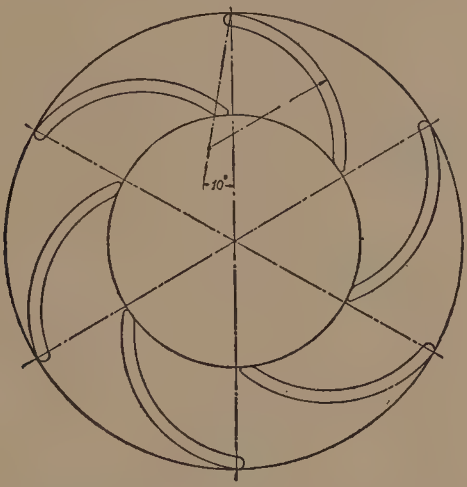


Fig. 397.—Theoretical Form of Impeller Vanes.

The pumps of automobile cooling systems have to work against a static head of only a few feet. For pumps working against such a small head the discharge angle of the vane should theoretically be 170°. That is, a tangent drawn to the vane curve at the outer end should make with a tangent to the impeller circumference at the same point an angle of 170°. Usually six vanes are used. The method of laying out the vanes is then as shown in Fig. 397. Most designers, however, regard this as an unnecessary degree of refinement, in view of the small amount of power involved, and make the vanes of plane form, either radial or cutting the tangent at a sharp angle.

Some designers extend the vanes all the way in to the hub, and this, of course, is necessary where they are not supported by a disc, but where a supporting disc is used it seems preferable to leave an open space of annular form between the hub and the vanes, the diameter of this open space being made equal to that of the water inlet. In some designs there are discs on both sides of the vanes, but this complicates the production of the impeller and is an unnecessary refinement.

Dimensions of Pumps—It is customary to drive the pump at crankshaft speed on four-cylinder and one and one-half times crankshaft speed on six-cylinder engines, though there are variations from these practices. An analysis of the dimensions of numerous pumps driven at crankshaft speed shows that the impeller diameter varies as the fourth root of the piston displacement of the engine. If we designate the piston displacement by V and the impeller diameter by D then

$$D = 1.1\sqrt[4]{V}.$$

The width of the vane at the impeller circumference is made $0.12D$, the diameter at the inner ends of the vanes and that of the water inlet are made $0.56\ D$. Theoretically the width of the impeller should increase from the circumference toward the center in such a way that the product of the radius into the width is constant, as that insures constant water velocity in a radial direction. This gives an hyperbolic outline, which is difficult to form. In practice the width of the vanes is decreased uniformly from the inner circumference to the outer. The width at the inner circumference is made $0.18D$.

It frequently happens that some other accessory is driven through the water pump and the proper diameter of the shaft then depends to some extent upon the power consumed by that accessory. In all ordinary cases, however, the shaft diameter may be made $0.15D$ and the impeller hub diameter $0.25D$.

Pump Dimensions for Other Speeds—If the pump is to be run at n times crankshaft speed, where n may be either greater or smaller than unity, then the impeller diameter should be made

$$D_1 = 1.1\sqrt[4]{\frac{V}{\sqrt{n}}}.$$

The other dimensions of the pump can be calculated by the same formula as for crankshaft speed.

Details of Construction—If some other accessory is to be driven through the pump shaft then the pump housing is often split centrally, the two halves being made symmetrical. A design of this type is shown in Fig. 398. This construction requires two bearings and two stuffing boxes on the shaft. Where no "through drive" is necessary it is much preferable to use only a single bearing, as there will then be only one place to keep lubricated and only one stuffing box to become leaky and require packing. In that case the pump housing proper, including the entire volute, is made in a single casting which is provided with a plain cover plate, preferably of sheet metal. The inlet may be either in the cover plate or around the pump bearing. Each bearing must be provided with lubricating means, preferably a screw-down grease cup.

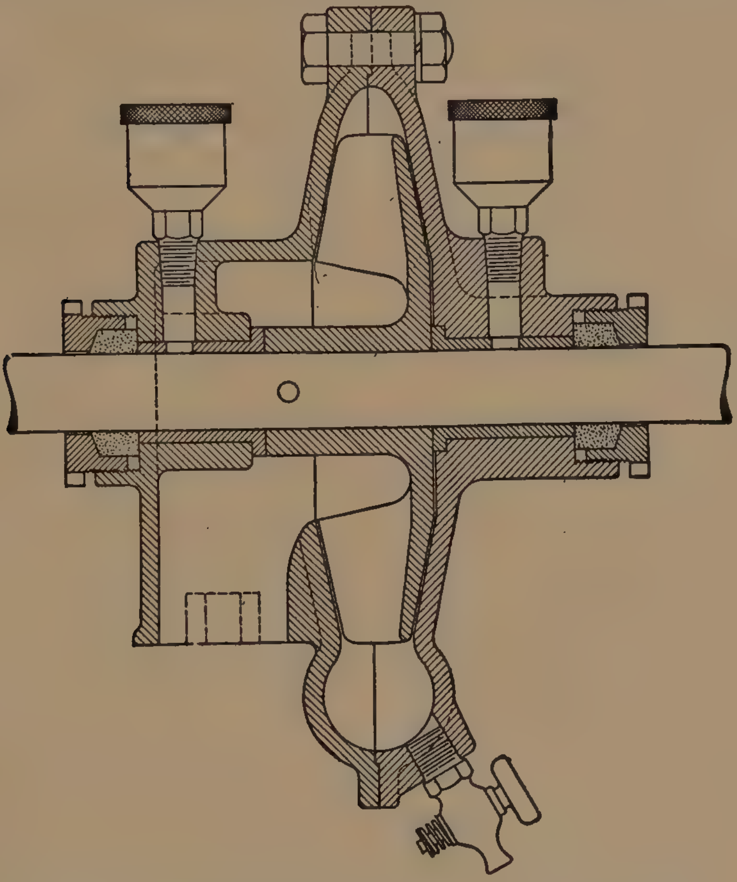


Fig. 398.—Typical Centrifugal Water Pump.

The volute, as already explained, increases uniformly in section all around the circumference. Its greatest section is equal to that of the water delivery pipe. Most designers make the section of the volute circular at every point, and this gives somewhat less resistance to the flow of the water than if the volute is made of uniform width and flattened to decrease the area. The latter scheme is occasionally used however, to decrease the radial dimensions of the pump.

The shaft of the pump is now often made of some rust-resisting alloy, such as monel metal or stainless steel. If ordinary steel is used the shaft is likely to rust up badly during periods of non-use, and then cuts its bearings and packing. The housings are made of cast iron, brass, aluminum or pressed steel, and the impellers of any of the cast materials mentioned.

As the water enters the pump from one side and leaves in a radial direction, there is a small end thrust on the propeller. This should be taken up on the bearing in the housing and the hub of the impeller, so that the impeller proper clears the housing all around.

The stuffing box is packed with candle wicking soaked in grease. A brown paper gasket stuck on with shellac dissolved in alcohol can be used between the two halves of the housing. A drain cock or plug should be placed at the lowest point of the housing, so that all water can be drained off in cold weather. Or, if the inlet fitting is cast integral with the pump housing the drain cock may be screwed into it and the pump chamber drained to the inlet fitting through a very small hole at its lowest point. Some of these details are illustrated in Fig. 399, which shows two views of the Stutz water pump. This pump is made with a flange with three lugs by which it is bolted to the rear of the timing gear case.

It will sometimes occur during the winter months that the owner forgets to drain the pump when leaving the car in a cold garage. The pump may then freeze up, and when it is attempted to crank the engine, it may be broken. In order that the damage done may be of such a nature as to be easily repaired, it is customary to insert a "weak link" either in the drive or in the pump itself. For instance, one member of the driving coupling may be secured to its shaft by a pin of such diameter as to shear off before any injury is done to the pump, or the impeller may be secured to the shaft by such a pin.

For V engines, in order to insure equal circulation of water through both cylinder sets, either two pumps are used, or else special double pumps with vanes on both sides of a central disc, two volutes and two outlets, which latter may be at different points of the circumference to suit the piping.

Combined Pump and Fan—In some cases the accessories drive may be materially simplified by mounting the water pump and fan on the same shaft and using one drive for both, and this practice is gaining ground rapidly. Figure 400 shows the arrangement on the Packard vertical eight. A chamber of considerable size is cast in the forward end of the cylinder block. Its front wall is bored out, and into it is set the combined pump housing and fan bracket, which is held in place by four studs. The common shaft runs in a bronze bearing at the pump end, with a stuffing box formed between the bearing bushing and a cap screwed over same, and in a flexible roller bearing at the fan end. Fan hub and impeller are secured to the shaft at opposite ends. The pulley for the V belt is formed integral with the fan hub. Belt adjustment is provided for by turning the flange on the pump housing eccentric to the shaft axis. Water inlet ports are formed in the pump housing cover plate on the rear. The inlet is cored below the chamber into which the pump is set and extends crosswise of the cylinder block, flush with the side thereof, an L-shaped, flanged fitting being bolted over its end.

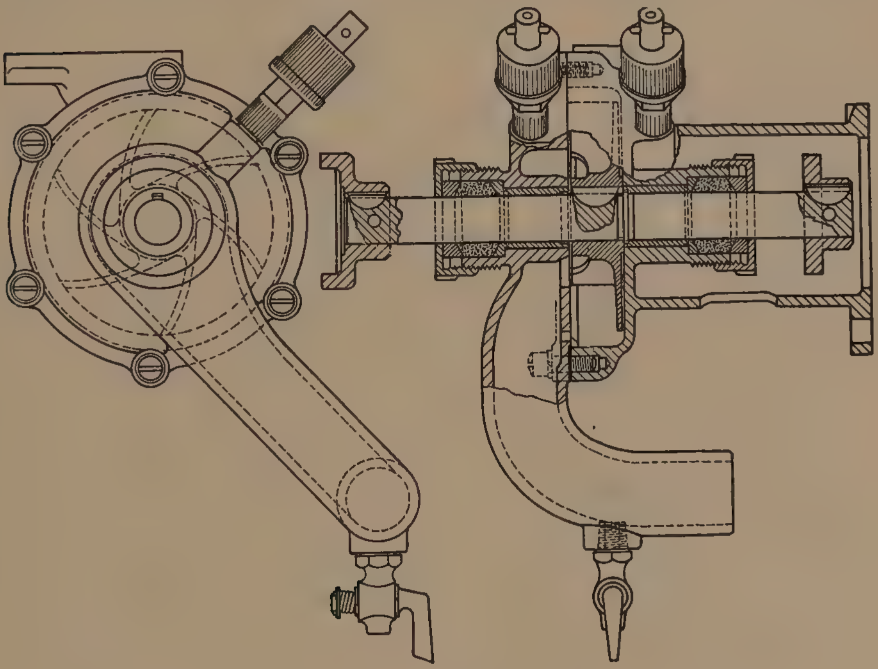


Fig. 399.—Stutz Design of Water Pump.

These combination pumps and fans came into use first in

England, and Fig. 401 shows a British design, the details of which differ considerably from those of the one previously described. The fan is a three-blade aluminum type; both fan and pump are supported by a bracket setting on top of the timing gear housing, which also forms the water inlet, and the pump housing sets right up against an opening in the front wall of the cylinder block, a water-tight joint being secured by means of a rubber ring of triangular section, forced into a similarly shaped groove at the joint by a hose clamp. The pump is not of the true centrifugal type but has vanes of helical form which move the water in the direction of the pump shaft. Such pumps are sometimes referred to as water circulators.

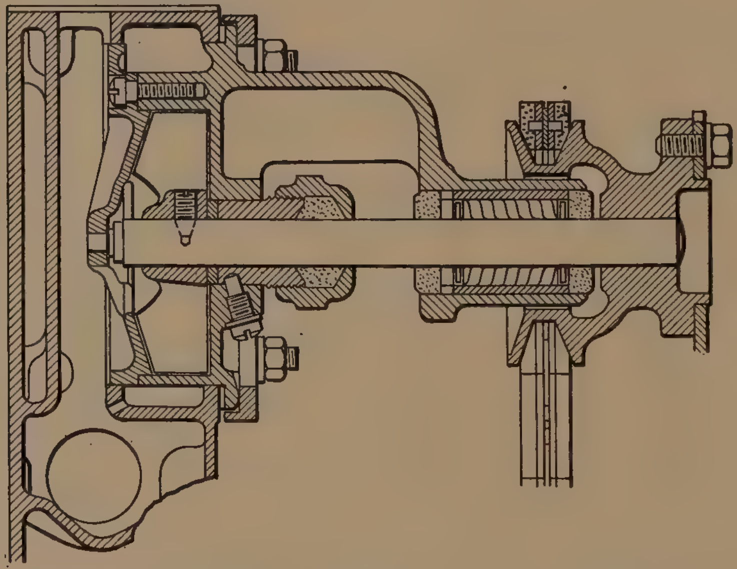


FIG. 400.—COMBINED PUMP AND FAN HUB (PACKARD).

Automatic Control of Circulation—Although the rate of water circulation and the rate of air delivery both increase and decrease with the speed of the engine, in the ordinary car the cooling effect does not at all conform to the changing requirements under different conditions of engine loading and atmospheric temperature. For instance, if the engine runs at a given speed, the water circulation and air circulation remain the same, whether the engine runs idle or under full load, and whether the atmospheric temperature is 100° or zero, though it is obvious that the heat to be dispersed is many times as great under full load as when idling, and the capacity of the radiator for dispersing heat is more than twice as great in zero weather as in hot summer weather. The result is that practically all cars which have a cooling system that is adequate under all conditions, over-cool their engines when lightly loaded and during cold weather, thus lowering the fuel economy and the ability of the engine to run smoothly at low speeds.

The cooling can be regulated by acting on either the water circulation or the air circulation through the radiator. By preventing the water from circulating through the radiator or by preventing air from passing through it as long as the jacket water is not up to a certain temperature, the engine can be quickly brought up to normal temperature when first starting from cold, and can be maintained at that temperature thereafter.

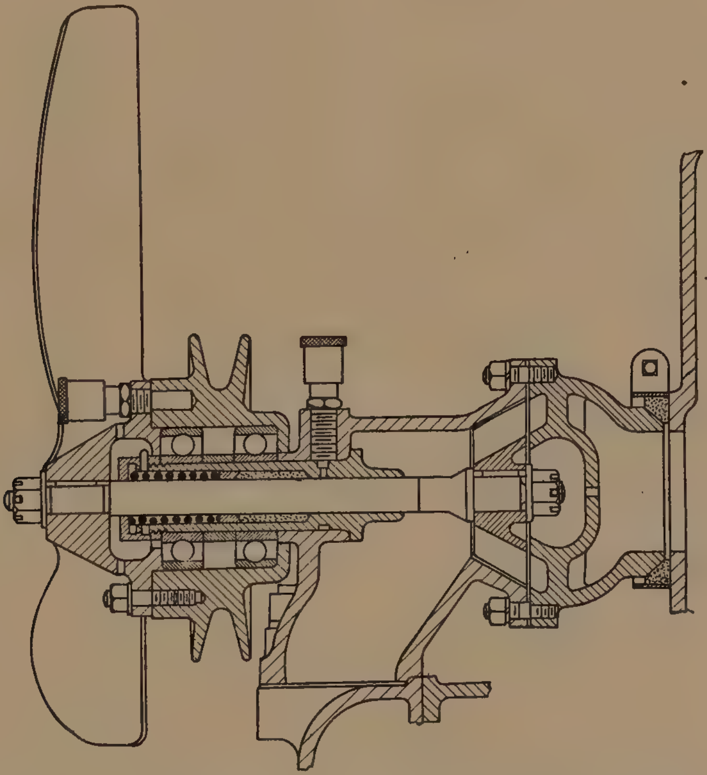


FIG. 401.—WATER CIRCULATOR COMBINED WITH FAN.

This effect is produced automatically by means of a vapor pressure type of thermostat. Thermostats were first applied to the control of water circulation, and this application is illustrated in Fig. 402. The thermostat consists of a "bellows," that is, a short length of circularly corrugated copper pipe closed at both ends and filled with a liquid of low boiling point (ether). This is so sensitive to variations in temperature that it expands visibly when merely breathed against. The copper thermostat is located in a chamber adjacent to the pump, and the valve is directly above it. Water from the radiator enters the valve chamber at the side, and there are two outlets for it from this chamber through the double valve, one upward and one downward, the object being to make the valve independent of the suction of the pump and subject only to the influence of the thermostat. There is a by-pass

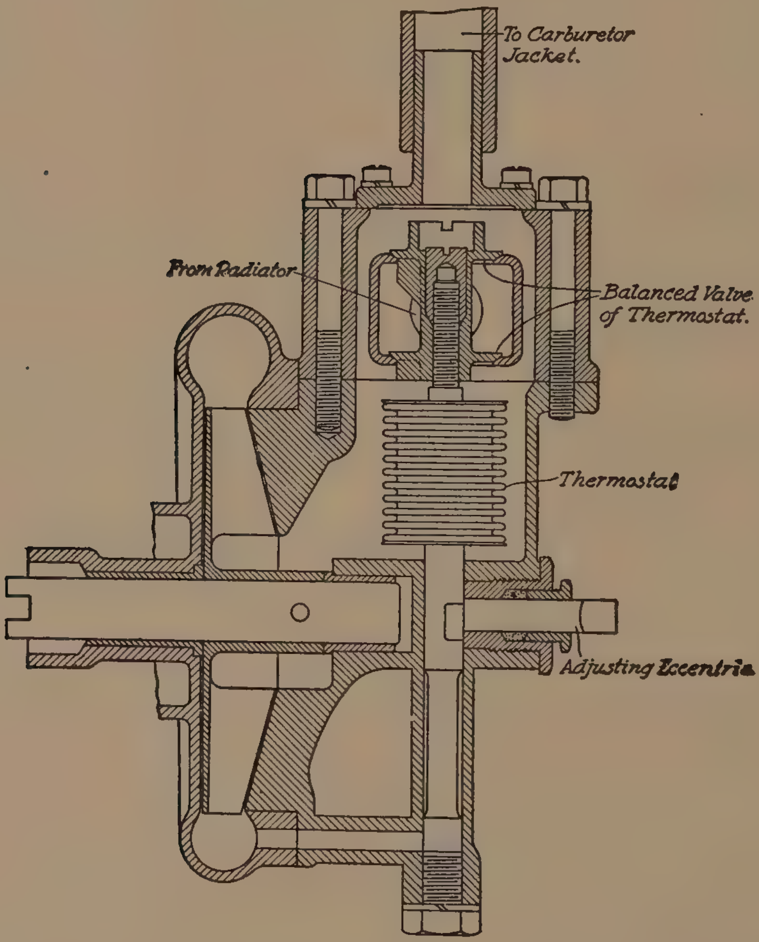


Fig. 402.—Thermostatic Control of Water Circulation.

from the thermostat chamber to the carburetor jacket which constantly remains in communication with the pump, so that water can circulate through the carburetor jacket all the time. As long as the water is cool the thermostatic valve is closed, and there is therefore no circulation except through the by-pass to the carburetor jacket. This insures quick warming up of the water in the engine jacket. As soon as warm water reaches the thermostat it expands and opens the valve, and then the circulation takes place in the regular way through the pump, radiator and jacket.

Figure 402 illustrates the first application of the thermostat, to the control of an automobile cooling system (on the Cadillac). In most of the later applications it was placed on the outlet from the engine jacket, instead of on the inlet line. Since in that case the hot water from the jacket naturally rises to the thermostat, the latter is probably somewhat more sensitive.

Automatic Radiator Shutters—An objection to automatic control of the water circulation is that with it there is considerable danger of freezing up the radiator in cold weather. If a driver starts out on a cold morning before his engine is up to its normal temperature, so that there is still no circulation through the radiator, a freeze-up may occur almost in an instant, because the radiator is filled with stagnant water already near the freezing point, in very thin layers, past which the cold air is forced on both sides, by both the motion of the vehicle and the action of the fan. Although nothing should happen if the system is filled with a non-freezing solution of the proper strength, some trouble has been experienced from this cause, and there is now a growing tendency to use automatic radiator shutters instead of automatic circulation control. These shutters are of the same general type as window shutters or blinds, but made of metal. They are actuated by a vapor-pressure type of thermostat as above described, which closes them whenever the temperature of the cooling water in the engine jacket is below a certain point, and vice versa. In the design of such shutters provisions must be made to prevent rusting at the joints, and the development of rattles.

Water Capacity—The average water capacity of a complete system for pump and thermo-siphon circulation, respectively, may be found by means of the following equations:

For pump systems

$$Q = 2 + \frac{C}{125} \text{ gallons;}$$

for thermo-siphon systems

$$Q = 3 + \frac{C}{125} \text{ gallons,}$$

where C is the piston displacement of the engine in cubic inches. The water capacity necessary with any system depends largely upon the type of radiator employed, being largest with the flanged tubular type of radiator and smallest with the air tube type in which all water spaces are "live." The rate of circulation also has an effect, as has the speed of the vehicle. The preceding equations apply to passenger cars only.

Evaporative Cooling—A great many small stationary internal combustion engines are cooled by the so-called hopper system, in which the heat absorbed by the jacket walls is dissipated by boiling off the water in the cooling system, a considerable amount of water being contained in a hopper-like reservoir cast integral with the engine cylinder. Recently efforts have been made to introduce similar systems—in which the steam is condensed, however—in automobile practice, and though evaporative cooling is not yet an accepted practice in the industry, there are indications that it will be given recognition before long.

The chief advantage of evaporative cooling is that, without the use of a thermostat, it maintains the cylinder wall at a constant temperature, and that the optimum temperature for systems employing water as a cooling medium. Except for a short time when first starting up, the engine jacket is kept at the temperature of boiling water, 212°, which should result in an appreciable increase in fuel economy. The jacket temperature is unaffected by atmospheric conditions.

Incidental advantages are that the higher cylinder wall temperature will aid the vaporization of the fuel and minimize crankcase dilution; that the system can be closed, so that there will be no loss of alcohol when a non-freezing alcohol solution is used, and that the radiator and (in some systems) the pump can be made smaller than with the conventional pump cooling system.

Rushmore System—An evaporative cooling system developed by S. W. Rushmore has the peculiarity that the mixture of hot water and steam which leaves the top of the jacket is conducted by a pipe to the bottom tank of the radiator. There the water is separated out by a baffle, while the steam rises in the core passages, and as it condenses, the water, in fine droplets, returns to the bottom tank. From there it is drawn by a pump similar (in type and size), to the gear type oil pump and forced back into the jacket. Mr. Rushmore also plans to use the steam and hot water for keeping the oil supply in the crankcase at a uniform temperature, and for steam-heating the interior of the car.

With an evaporative system of this kind the rate of circulation is naturally much smaller than with the ordinary system, for if the temperature drop in the radiator of the ordinary system is 20°, each pound of water passing to the radiator carries with it 20 B.T.U. which it does not take back to the jacket, whereas one pound of water passing to the radiator in the form of steam, carries with it 972 B.T.U. which it will not take back if it returns as water. Hence the rate of circulation need be only about one-fiftieth as great with the evaporative cooling system. The steam will rise in the radiator to a height determined by the rate of heat transfer to it. An air valve is placed on the upper part of the radiator, and a blow-off valve on the pipe through which the steam passes to the radiator, at the highest point of this pipe, which is set to blow off at 1 or 2 lbs. per sq. in. A diagram of the Rushmore system is shown in Fig. 403.

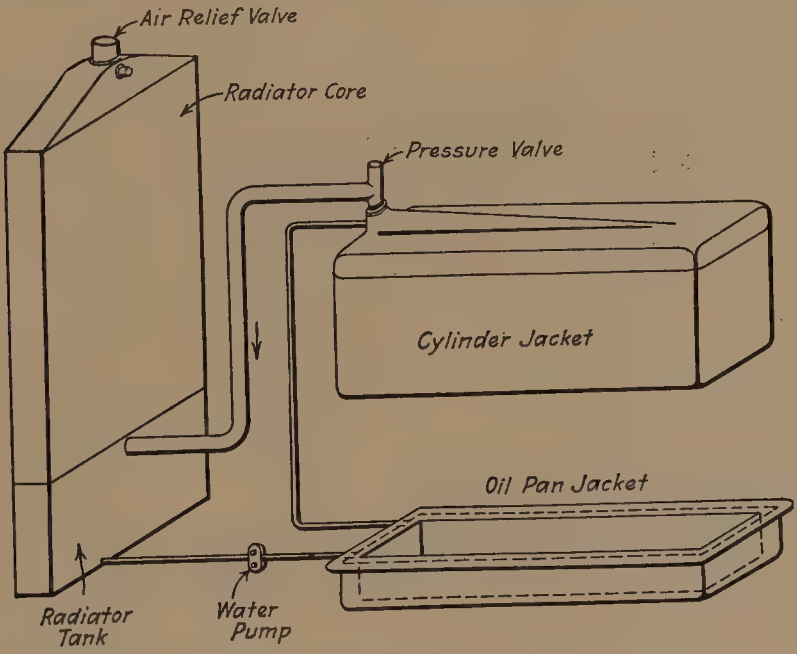


FIG. 403.—RUSHMORE EVAPORATIVE COOLING SYSTEM.

Another evaporative cooling system, known as the Iso-

Therm and made by the Harrison Radiator Co., is illustrated in Fig. 404. In this system the passages in the radiator core are horizontal rather than vertical, and there is a tank at each side. The outlet from the engine jacket connects by a hose to an inlet to the tank at the left side, near the top, and from the bottom of the tank on the right a hose connection extends to the pump inlet. In addition, a small tube leads from the pump intake to the upper part of the right tank, its purpose being to release any air which may be trapped at the intake.

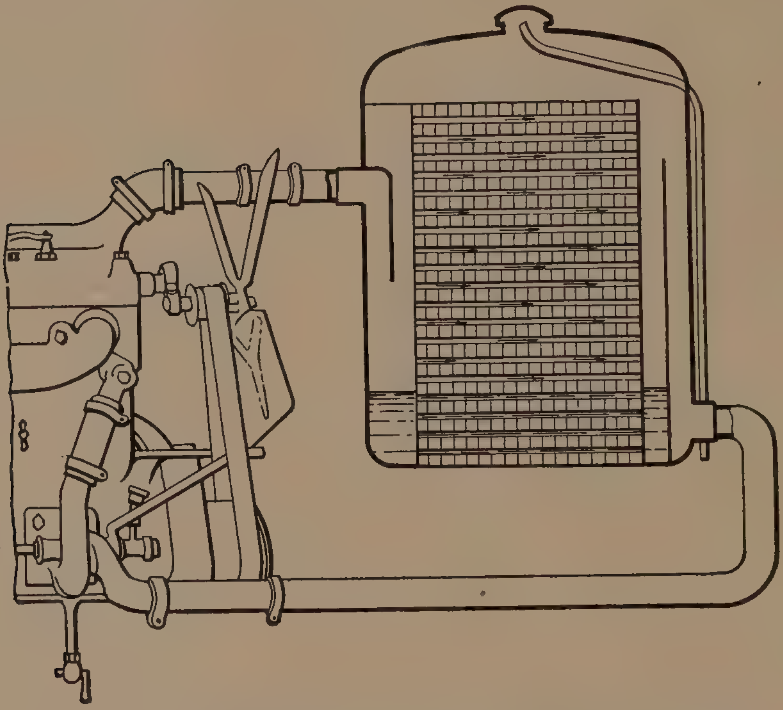


Fig. 404.—Iso-Therm Evaporative Cooling System.

Petcocks are screwed into one of the tanks of the radiator to mark the maximum and minimum water levels, and when the system is filled, water is poured in until it starts to run out of the upper cock.

The steam generated in the jacket, together with a certain amount of water held in suspension, enters the radiator near the top of the left tank. It there strikes a vertical baffle plate which tends to separate out any entrained water. The relative amount of water passing over with the steam depends upon the engine load, being greatest at full load.

Water separated from the steam by the baffle drops to the bottom of the tank and passes over to the right side of the radiator through the lower passages of the core, while the steam passes over through the upper passages. All water collects at the bottom of the right tank, whence it is drawn by the pump. The vent is located on the cold side of the radiator.

With one design of the Iso-Therm system a small auxiliary condenser is placed on top of the left-hand tank. This is intended to take care of the steam formed when the engine is stopped after having run under full load for some time, due to the equalization of the temperatures of the cylinder walls and the water. Since there is then no air passing through the core, its ability to dissipate heat is greatly reduced, and if it were not for the auxiliary condenser some of the cooling fluid would be lost through the vent.

CHAPTER XXIV

Air Cooling

The heat absorbed by the cylinder walls of any explosion engine is finally dispersed in the atmosphere. With the water-cooling system the water merely constitutes a vehicle for transferring it from the engine to the radiator, which latter gives it off to the atmosphere. With air cooling the heat passes from the cylinder walls directly to the atmosphere, and advocates of air cooling therefore refer to it as the direct cooling system.

Air cooling was used by Gottlieb Daimler in his first bicycle engine, but was soon discarded by him. It was successfully applied to cycle engines by Bouton about 1897. These engines had single cylinders of very small bore and stroke. It is easily proven that a small cylinder can be more easily kept cool than a large one. With the same temperature of cylinder wall the amount of heat radiated per unit of wall area will be the same in cylinders of all sizes. But since in a large cylinder the wall area is smaller, in proportion to the combustion chamber volume, more heat will have to be absorbed per unit of wall surface in order that the burning gases may drop in temperature as rapidly as in the small cylinder. Moreover, in some air-cooled cycle or vehicle engines one side of the cylinder is exposed to a current of air, due either to the motion of the vehicle or to a fan, while the other side is shielded from this draught, and this shielding is the more effective the larger the cylinder dimensions. Consequently, the tendency to overheat is the greater the larger the cylinders.

Cooling Flanges—The amount of heat which will be absorbed or radiated by a unit of metallic surface depends upon the temperature difference between the surface and the surrounding air. If the combustion chamber wall had the same area of surface in contact with the hot gases on the inside and the atmospheric air on the outside, then it would tend to assume a temperature which would be a mean between the atmospheric temperature and the average temperature inside the cylinder, particularly if the conditions of motion of the gaseous elements inside and outside the cylinder, with respect to the cylinder walls, were about the same. But this temperature would be far too high to permit of satisfactory lubrication, not to mention other likely troubles. According to some experiments made by Prof. H. L. Callender, the operating conditions of an air-cooled engine cease to be satisfactory when the cylinder head attains a temperature of 570° F., but others have worked successfully with head temperatures of over 600°.

Reduction of Heat Absorption—In order to keep down the cylinder wall temperature it is necessary to so shape and finish the inner surfaces that they will absorb a minimum amount of heat, and the outer surfaces so that they will radiate this heat most readily. It has been found by laboratory experiments that a polished metal surface does not radiate heat nearly as rapidly as a rough surface, and it is logical to assume that when in contact with hot gases it will not absorb as much heat either. It is therefore advisable to give a smooth machine finish to the whole interior surface of the combustion chamber. This can only be done if the valves are located in the cylinder head, and this valve arrangement is therefore preferred for air-cooled engines. There is one other advantage in polishing the combustion chamber wall surface. It is found that spontaneous ignition or pre-ignition in both water- and air-cooled engines is often caused by carbon deposits on the cylinder walls. Carbon is a poor conductor of heat and attains a very high temperature when adhering to the combustion chamber wall. When the wall surface is polished the solid carbon formed by imperfect combustion will not so readily adhere to it, but will be blown out with the exhaust.

The temperatures of explosion and combustion depend directly upon the compression pressure. In an air-cooled engine the compression, therefore, must be kept fairly low. A compression ratio of 4 to 1 is about the practical limit for air-cooled engines, while a ratio of 5 to 1 is occasionally used in water-cooled engines. Rich mixtures tend to heat the engine more than mixtures properly proportioned or somewhat lean, and it is, therefore, advisable to so arrange the carburetor that it is impossible for the driver to adjust it so as to make the mixture over-rich. Quick evacuation of the cylinder at the end of the power stroke is also essential, and large size, quick-opening exhaust valves must therefore be used.

Radiating Surface—The combustion chamber wall disperses the heat which it absorbs from the burning gases almost entirely by convection, though there is also a slight dissipation of heat by radiation. Convection from any surface is substantially proportional to the speed at which the air moves over it. Both radiation and convection depend upon the area of the surface exposed and to some extent upon the condition of this surface. It is therefore necessary, in order to effectively cool the combustion chamber, to make the outer surface of the cylinder wall as large as possible and to pass cool air over it at a rapid rate.

Form and Dimensions of Cooling Fins—In very small engines, such as cycle engines, the only provision made for insuring effective cooling consists in increasing the radiating surface by providing the cylinder with cooling fins. The planes of these fins should always be in the direction of air flow.

There are two forms of air cooling, known, respectively, as free air cooling and blower cooling. Free air cooling can be used on large aircraft engines, particularly of the rotary and radial type, but in automobile work its field is limited. In fact, its use is confined almost entirely to motorcycle engines. Instead of casting cooling fins on the cylinder casting, sheet metal fins may be set in the mold and have the cylinder cast onto them, and this practice is followed particularly in connection with blower cooling.

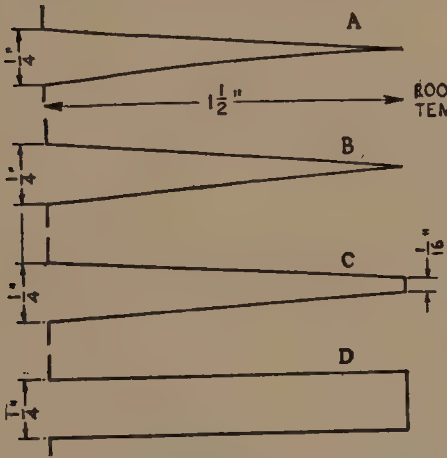
A theoretical and experimental analyses of air cooling was made by C. B. Dicksee, of the Westinghouse Electric & Mfg. Co., who published a summary of his researches on the subject in *Automotive Industries* of April 5, 1923.

Heat Dissipation from Fins—The heat given off to the atmosphere by a hot surface, like that of a cooling fin, is directly proportional to the difference between the temperatures of the fin and the surrounding air. The temperature of the fin, however, decreases from its root to its tip, because it is this temperature gradient which causes the heat to flow from the root to the tip. Hence, since we may assume the air temperature to be the same over the whole depth of the fin, it follows that the fin surface near the root gives off heat at a higher rate and is therefore more efficient.

The efficiency of the outer portion of the fin depends upon the facility with which heat can reach it. This, in turn, depends upon its distance from the root, the cross-sectional area for the heat to flow through and the heat conductivity of the material. The same as in a mechanical structure, the efficiency of utilization of fin material is a maximum when all parts are equally stressed.

In a cooling fin the material is used to the best advantage if the drop in temperature from the root to the tip is constant or uniform. This consideration leads to a razor-blade section, which, however, is not practical for other reasons.

Four different forms of fin are illustrated in Fig. 405 reproduced from Mr. Dicksee's article, and the drop in temperature from root to tip with each type is shown by a curve in Fig. 406. The vertical height of the curves in Fig. 406 represents the difference between the temperature of the atmosphere and the temperature of the particular part of the fin, and is, therefore, a measure of the relative effectiveness of different parts of the fin.



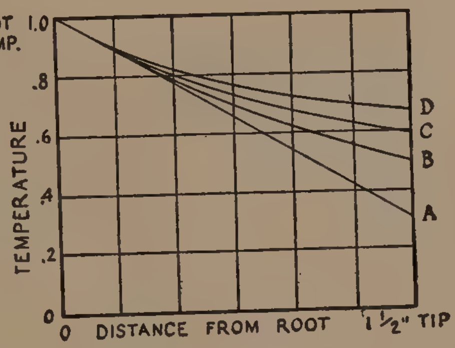


Fig. 405 (left).—Four Forms of Cooling Fin.
Fig. 406 (right).—Temperature Gradients of the Four Fins.

It was stated above that the rate at which heat is dissipated by the fins is substantially proportional to the velocity of the air moving past them. Accurate measurements have shown that the heat dissipation varies slightly less rapidly, as the 0.89th instead of as the first power of the air velocity, and is equal to 0.066 $V^{0.89}$ B.T.U. per square foot per minute, per degree Fahrenheit, where V is the air velocity in thousands of feet per minute. This would make it very easy to calculate the heat-dissipating capacity of a given fin area were it not for the fact that the temperature of the fin necessarily always varies from point to point.

Sheet metal fins are always quite thin, and, owing to the limited heat conductivity of these thin sections, there is no use in making them very deep, as the outer portions would not get enough heat. Starting with a fin of very small depth, the heat dissipating capacity increases at first almost in direct proportion to the depth, but then a point is reached where there is a decided lessening in the rate of increase, a point similar to the saturation point of the magnetic induction curve. However, the depth corresponding to this saturation point with material of any given thickness varies with the air velocity—the higher the air velocity the smaller the depth of fin beyond which it is inadvisable to go.

Mr. Dicksee carried through some rather elaborate calculations to determine the heat-dissipating capacities of fins of different thickness as affected by air velocity and depth of fin. His results were given in the form of curves. In the following table are given what appear to the author to be the optimum depths of steel fins for four different air speeds and four thicknesses, and their respective heat-dissipating capacities in B.T.U. per minute per inch in length per degree F. temperature difference between the root of the fin and the atmosphere.

Table XV—Heat Dissipated by Sheet Steel Fins per Minute, per Inch in Length, per Degree F. Temperature Difference

Thickness of Fin	Air Velocity Ft. per Min.	Optimum Depth (In.)	B.T.U. per Inch Per Min. per Deg. F.
0.0312	2500	1¼	0.00143
0.0500	2500	1½	0.00180
0.0625	2500	1⅞	0.00215
0.0312	4000	⅞	0.00164
0.0500	4000	1¼	0.00209
0.0625	4000	1½	0.00243
0.0312	6800	⅝	0.00196
0.0500	6800	⅞	0.00253
0.0625	6800	1	0.00288
0.0100	6800	1⅜	0.00343
0.0312	9000	½	0.00206
0.0500	9000	¾	0.00279
0.0625	9000	1	0.00326

Even with the thinnest fins it is not practicable to set them closer than five to the inch, as if too close the spaces are likely to become clogged with insects, dried foliage, etc.

Cast Fins—Cast fins must be made with a taper of from 5° to 7°, the latter about equal to ⅛ inch per inch, in order that the pattern may withdraw readily from the mold. A thickness of 1/16 in. at the tip, with a taper of ⅛ in. per inch or very slightly less, seems to give the best results. Fig. 407 shows the cylinder of the J. A. P. engine, one of the most successful British motorcycle engines. It has a bore of 74 mm. (nearly 3 in.) and a stroke of 80 mm. (3.15 in.). The fins at the upper end of the cylinder are 1¼ in. high, those at the lower end of the stroke a little less. The spacing is 5/16 in. and the thickness appears to vary from about 3/16 in. at the root to 1/16 in. at the tip.' On the detachable head the fins are 1½ in. high.

Blower Cooling—It is very difficult to successfully apply the free air-cooling system to multi-cylinder engines arranged lengthwise under the hood, and it is certainly impossible to obtain a high output from engines so cooled. For such engines the blower cooling system offers advantages. This system has been successfully used by the H. H. Franklin Mfg. Co., for many years. An illustration of the Franklin cylinder is shown in Fig. 408. It has a bore of 3¼ in. and is provided with 104 copper fins 0.025 in. thick and 1 in. deep, the outer 3/16 in. being bent at right angles to form a jacket. A side elevation of the Franklin engine is shown in Fig. 409, which shows a centrifugal blower located at the front end, with its impeller directly on an extension of the crankshaft. The air

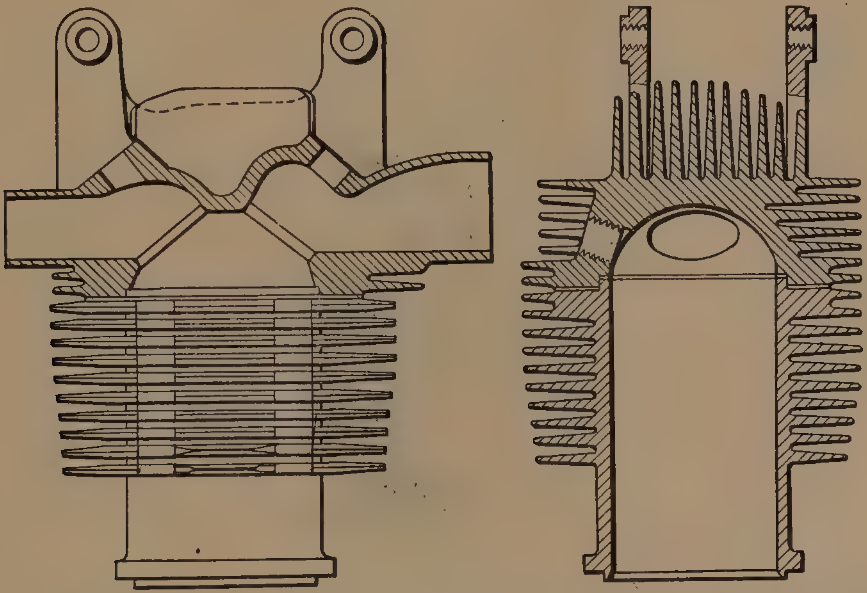


Fig. 407.—Air-Cooled Cylinder of J. A. P. Motorcycle Engine.

is collected in a volute and then led through a duct of rectangular cross-section, which extends the length of the engine on top of the cylinders and makes an air-tight connection with the cooling jackets of same. This large duct can be clearly seen in the sectional view of the engine, Fig. 410. Baffles are arranged in the duct over each cylinder, of such form as to produce a substantially equal division of air flow between the different jackets. The cool air first strikes the valves, which have no cooling fins, and it then passes down the jackets and escapes.

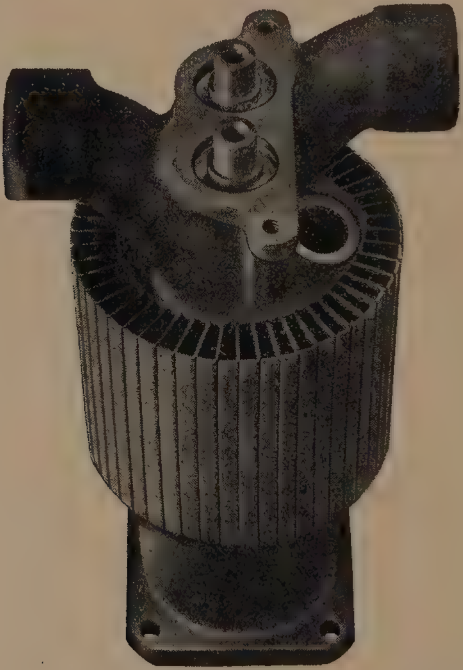


Fig. 408.—Franklin Air-Jacketed Cylinder.

The Blower—A centrifugal fan or blower consists of two parts, viz., the casing, whose peripheral surface is of spiral shape, and the runner. The latter is a revolving wheel with numerous blades or vanes on its circumference which cause the air to revolve with it. The air enters through the open side of the casing (see Fig. 411), has a rotary motion imparted to it by the blades, and is forced outward against the casing spiral by its centrifugal force, where its velocity is partly converted into pressure, which pressure forces it through the duct and jacket.

Figure 411 shows the blower used on the Frayer-Miller engine, which was probably the first to use this system of cooling. This blower was located with its axis above the crankshaft axis and was driven from the crankshaft by belt. Since the view shown is that from the front of the car, the impeller rotates left-handedly and the vanes therefore are tipped backward. The same laws which apply to a centrifugal water pump and which were discussed in the preceding chapter also apply to a centrifugal air pump. By inclining the blades backward a low air speed in the pump, and consequently a low air pressure, is produced. Owing to the fact that air is very light, the pressure is naturally quite low. On the other hand, the narrow passages in the air jackets of the cylinders offer considerable resistance to the flow of the air, and in some recent designs it has been found advantageous to tip the blades forward to increase the pressure.

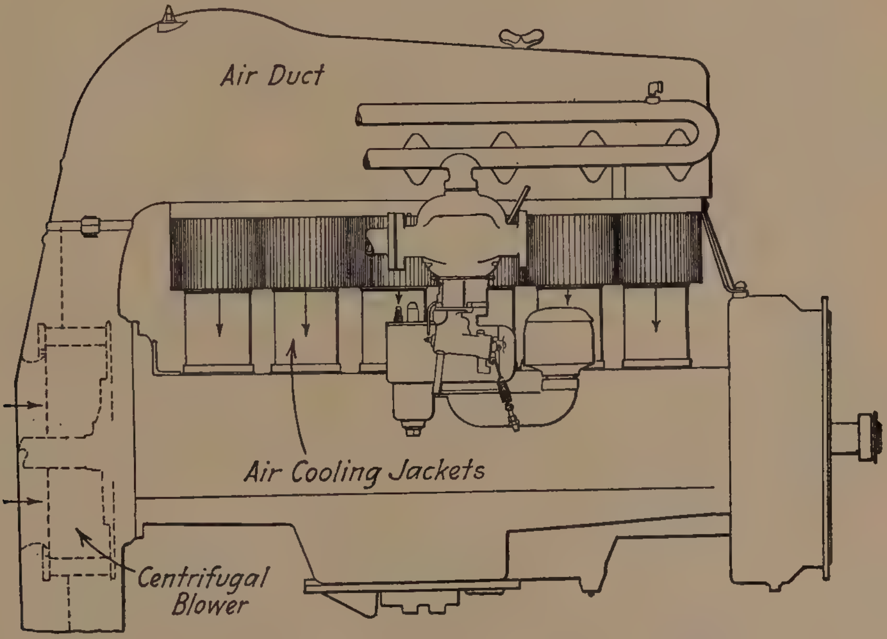


Fig. 409.—Side View of Franklin Engine, Showing Blower, Air Duct and Cylinder Jackets.

Air Circulation Necessary—Suppose that a 25-hp. engine is to be cooled by the blower system. We may assume that the fuel consumption is 0.65 lb. of gasoline per hp.-hr., and that 40 per cent of the fuel energy is carried off by the cooling air. What must be the delivery of the blower at the full

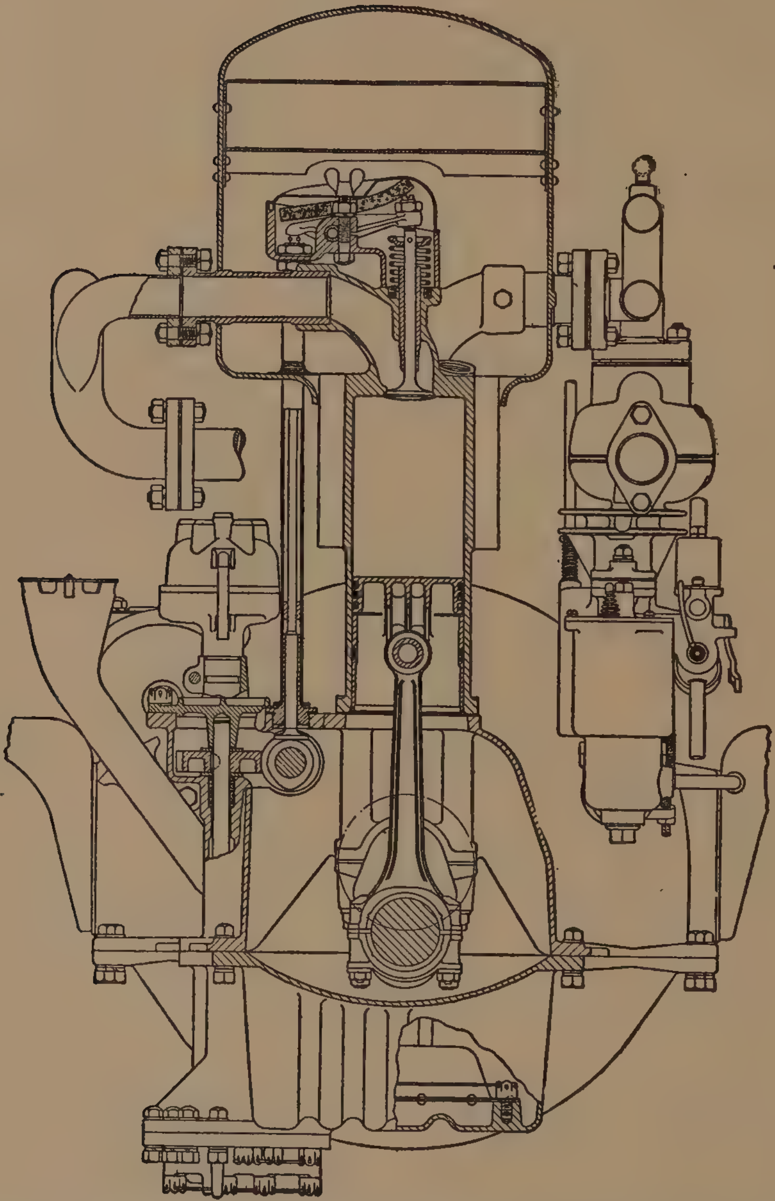


FIG. 410.—SECTION THROUGH FRANKLIN ENGINE, SHOWING AIR DUCT AND BAFFLES.

load speed of the engine? The heat energy to be carried off by the cooling air is evidently

$$\frac{0.65 \times 25 \times 19{,}000 \times 0.40}{60} = 2060 \text{ B.T.U. (appr.) per min.}$$

Now, the specific heat of air at constant pressure is 0.24. Allowing for a rise of 100° F. in the temperature of the air, the amount to be moved is

$$\frac{2060}{100 \times 0.24} = 86 \text{ lbs.}$$

Since atmospheric air weighs 0.077 lb. per cu. ft. this is equal to

$$\frac{86}{0.077} = \text{practically } 1120 \text{ cu. ft. per min.}$$

At full load, therefore, a circulation of 1120 cu. ft. per min. would limit the rise in temperature of the cooling air to 100°. Cooling conditions, however, are more severe when the engine is pulled down in speed with the throttle wide open. At one-half the normal full-load speed, for instance, the torque will probably be about 25 per cent greater, hence the horse power output will be 62.5 per cent of the maximum, whereas the air delivery, which is directly proportional to the speed, will be only 50 per cent of that at full load. Hence, if it is desired to limit the rise in temperature of the cooling air to 100° under the worst conditions, the rate of air circulation must be increased in the ratio of 62.5 to 50 or by 25 per cent. This would make the rate of circulation at full load:

$$1.25 \times 1120 = 1400 \text{ cu. ft. per min.,}$$

which is about right.

Cylinder Wall Temperatures—Figure 412 is a plot of temperatures measured around the edge of the cylinder head of the Franklin 9-B engine, at points indicated in the subfigure, by means of thermo-couples. All readings were taken on cylinder No. 2. The throttle was kept wide open at the different speeds, and the horse power outputs were as follows: 600 r.p.m., 12.24 hp.; 1800 r.p.m., 25.4 hp.; 2200 r.p.m., 22.8 hp. The cylinder compression was 54.5 lbs. per sq. in. at 500 r.p.m. and 52.5 lbs. at 1200 r.p.m.

As would be expected, the temperatures all around increase and decrease with the output. It will be noticed that under full load the circumference of the head attains a temperature of about 650° on the exhaust side, which is about 300° higher than the temperature on the opposite side under the same conditions. This seems to be about the permissible limit.

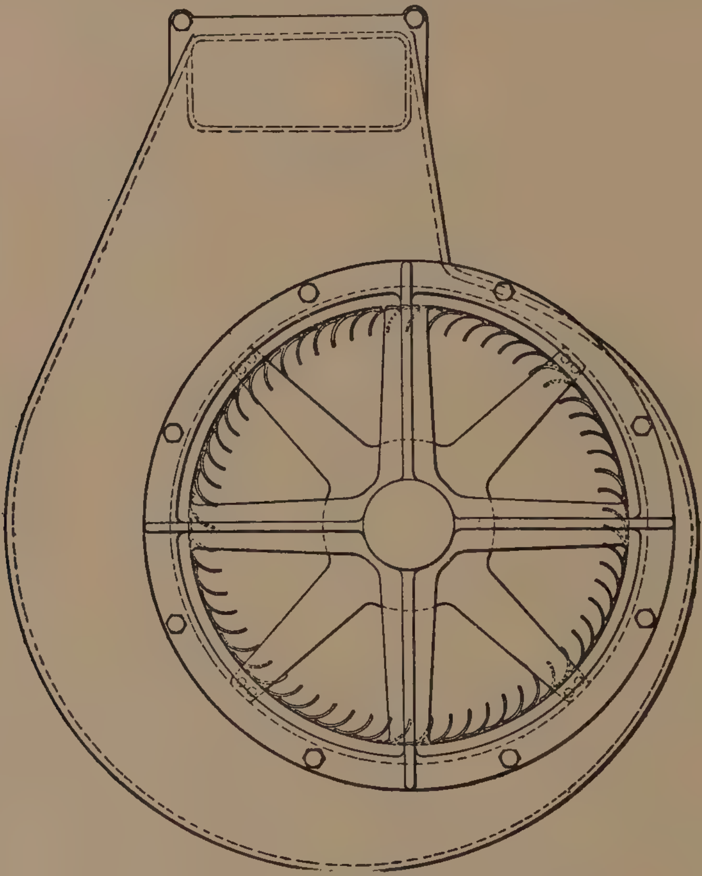


Fig. 411.—Centrifugal Blower.

Temperature measurements were also made down the side of the cylinder. In the center plane of the engine, at 1600 r.p.m. when developing 25.8 hp., the temperature was 620° 1 in. down from the top of the cylinder head; 450° 3 in. down; 425° 5.5 in. down and 350° 9 in. down. At 1000 r.p.m. when developing 18.1 hp., the temperature was 536° 1 in. down; 395° 3 in. down; 375° 5.5 in. down and 310° 9 in. down. At 600 r.p.m., when developing 12.2 hp., the temperature was 430° 1 in. down; 320° 3 in. down, 305° 5.5 in.

down and 250° 9 in. down. Under all load conditions the temperature drops rapidly over the first few inches from the top, then remains almost stationary to the end of the fins, and then drops again fairly rapidly to the base flange of the cylinder.

General Points on Air Cooling—Since the dissipation of heat from fins varies almost in direct proportion to the velocity of the air past them, it would seem that air cooling should impose no limits on the power obtainable from a cylinder of a given size, as all that would be necessary to carry off any additional heat absorbed by the cylinder walls if the specific output were increased, would be to increase the velocity of the cooling air by enlarging the capacity of the fan. However, when the air speed through the jacket is increased, the power absorbed by the fan increases at a much more rapid rate, hence the loss in the fan may exceed the gain in the cylinder. Within reasonable limits the air moved by a blower varies as the first power of the speed of the

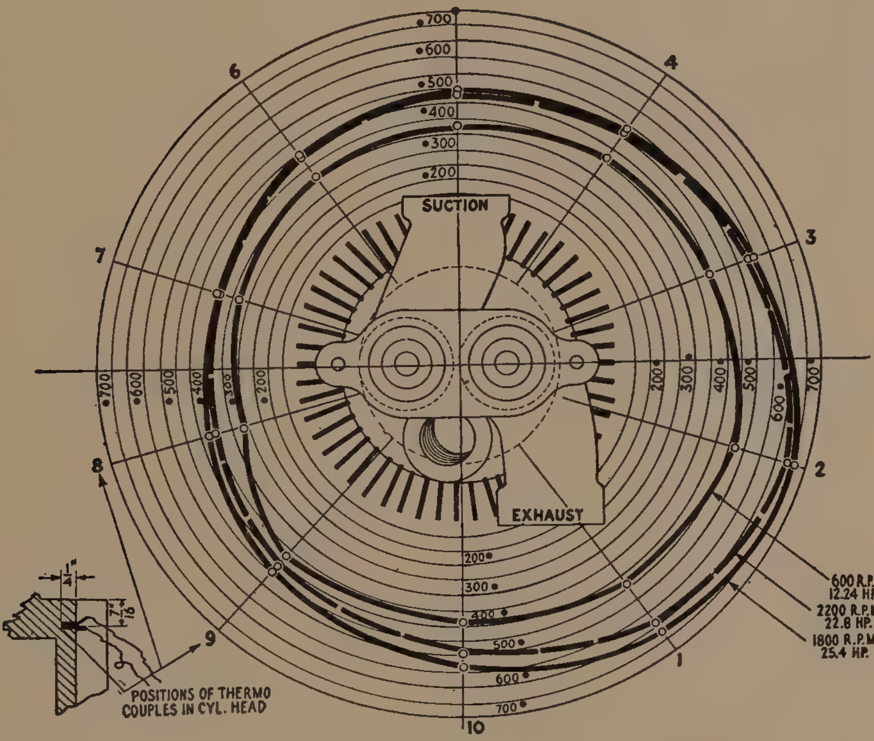


FIG. 412.—CYLINDER-HEAD-TEMPERATURE CHART TAKEN FROM FRANKLIN ENGINE.

blower; the pressure of the air at the outlet from the blower varies as the square of that speed, and the horse power consumed in driving the blower, as the cube of the speed.

To be able to increase the specific output from an engine cylinder, the blower, duct and jackets should all be so designed as to ensure the easiest possible flow of the air. If the efficiency of the blower were calculated on the basis of the power represented by the jet of air at its maximum speed, it would be found to be very low, which is due to the fact that there are abrupt changes in section and direction in the air channel, which give rise to eddies and consequent losses.

The blast should always be directed against the valves first (the exhaust valve purposely and the inlet incidentally), even though this involves forcing it down the jacket, in opposition to its natural tendency to rise, for if air were drawn up the jacket, by the time it reached the valves it would have become quite hot and its cooling capacity would have been much reduced. Besides, valves in the cylinder head are generally quite well shielded against a blast of air coming up through the jacket. Moreover, the air intake should always be located forward of the points of discharge, otherwise more or less of the heated air will be drawn back into the system and impair the cooling effect.

CHAPTER XXV

Assembling, Running-in and Routine Tests

After the various parts of an engine have been completed, they must be assembled, and then the completed engine is "run-in" and tested. In modern plants, parts production and assembling go on simultaneously, in different departments. Assembling is done largely on the movable line or conveyor plan. Starting with the crankcase or cylinder block, the partly assembled engine moves down a line as one part after another is put in place. Certain sub-assemblies, such as the crankshaft, flywheel and cam gear pinion, are put together in another part of the shop, and these sub-assemblies, together with other individual parts are delivered at points along the main assembly line where they are put on, sometimes along what are known as spurs to the main assembly line.

The whole idea in modern-assembly layouts is to relieve the men of all work that can be done by power. In some cases the engine frame is moved along mechanically while assembling is going on, while in others the blocks are secured to racks that move on rollers on rails. Use is also made of pneumatic and electric hoists for handling the blocks, and the racks are often of the so-called universal type, permitting the block to be turned or tipped in any direction, so that all assembling operations can be performed most conveniently. Following is an outline of the assembling operations on a small six-cylinder engine which has the cylinders and crankcase cast in a single block.

Assembling Operations—After the camshaft bearings have been forced into position in an arbor press, the inside of the crankcase is given a coating of a red enamel in which an alcohol dryer is used to cause it to dry quickly. Next the valve seats are turned with a compressed air tool, a roughing and a finishing operation being used. A valve is then fitted to each seat and ground in, and each valve is marked so it can always be returned to the proper place. The grinding-in operation is performed in a multi-spindle machine.

To protect the valves and their seats against injury in succeeding operations, and especially while the block is being slid along the conveyor, a dummy cylinder head is put on. The block is then turned upside down, the main bearings are put in, and these and the camshaft bearings are line-reamed with Martell reamers—a hand operation. Reaming is in two stages—roughing and finishing—and after this is done the chips are blown away by means of an air blast. Plug gauges are used to check bearing diameters, not necessarily on all engines coming down the line, but on a sufficient proportion to guard against inadmissible variations due to wear of the reamers.

After the flywheel housing has been bolted on, it is bored and faced true with the transmission shaft, these operations being located from dowel holes in the front and rear bearings. Next the pistons are put into the cylinders, and in this case what is known as selective assembly is used. That is, for each cylinder a piston is selected by trial which will just permit of the insertion of a 0.004-in. feeler between it and the cylinder wall. Now the crankshaft is put in place and the main bearing caps are bolted on.

Next the engine is put on a conveyor arranged in the form of a closed circuit. As it moves along, the pump assembly is bolted on and the pistons with their connecting rods, pins and rings, each forming a sub-assembly, are put in. These sub-assemblies arrive via a spur to the main assembly line. Considerable pains are taken with this sub-assembly. The piston pins, which must be a very accurate fit because of the high bearing pressure on them, are assembled selectively, a very small clearance being allowed. The bearings are put in place in the connecting rods and line-reamed, a gauge being used to ensure that the axes of the two bearings are in the same plane. Gaps in piston rings (when compressed to the diameter of the bore) are checked, the limits being 0.004 and 0.010 in. Pistons are micrometered for roundness, and before they are put into the cylinder they are thoroughly cleaned in gasoline and then flushed in oil.

Belting-in—All of the major parts being now in place, the engine goes to the "belting-in" stand. "Belting in" or "running in" is necessary in order to wear down the high spots on all bearing surfaces and thus reduce the friction. It is done either by belt from a line shaft or by individual electric motors. In the former case a belt pulley is secured to the flywheel and the engine is turned over for an hour or so to limber it up. This operation completed, the pulley is removed and the bolts holding the caps of the main bearings in place are locked by wiring. Next the dummy head is removed and the engine placed on a conveyor which takes it to the final assembly line—a traveling conveyor carrying assembling racks. These racks are of the universal type so the workman can put on parts on top and below with equal facility. Before assembling operations on this line start, the block is once more cleaned with gasoline.

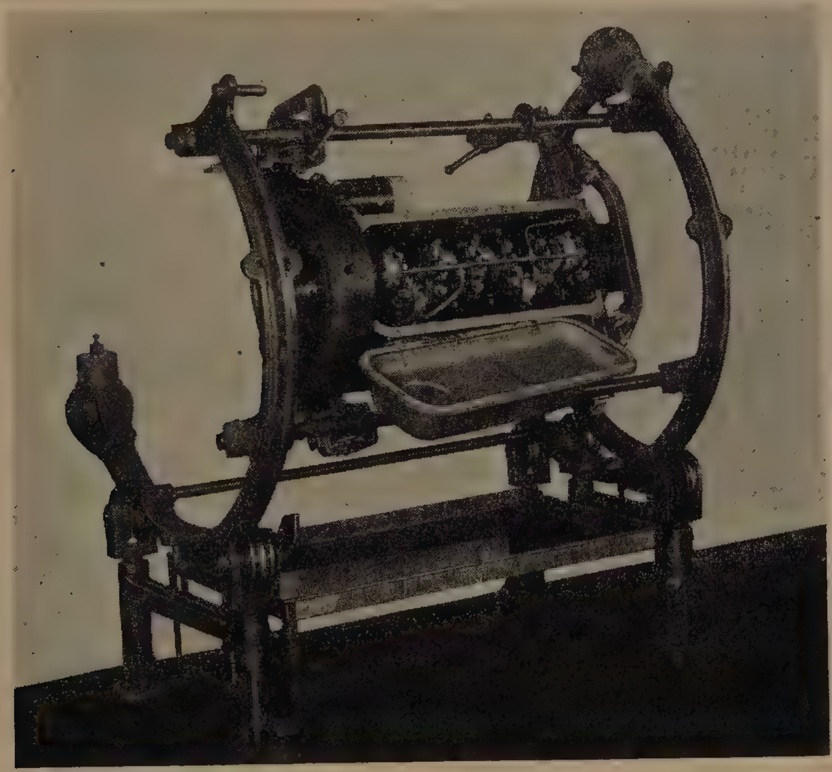


Fig. 413.—Continental Tip-Up Engine Stand, Similar to that Built into Moving Assembling Line.

Timing gears—all except the crankshaft pinion—are put in place while the block is tilted, with the front end up. While the valve tappets—which are held in clusters—are put in, the engine is turned at an angle sideways. Next the various studs, such as those for the cylinder head, manifolds, etc., are set, and for this operation power tools are now often used. After the regular cylinder head is put on and while the engine moves down the line, such smaller parts as the

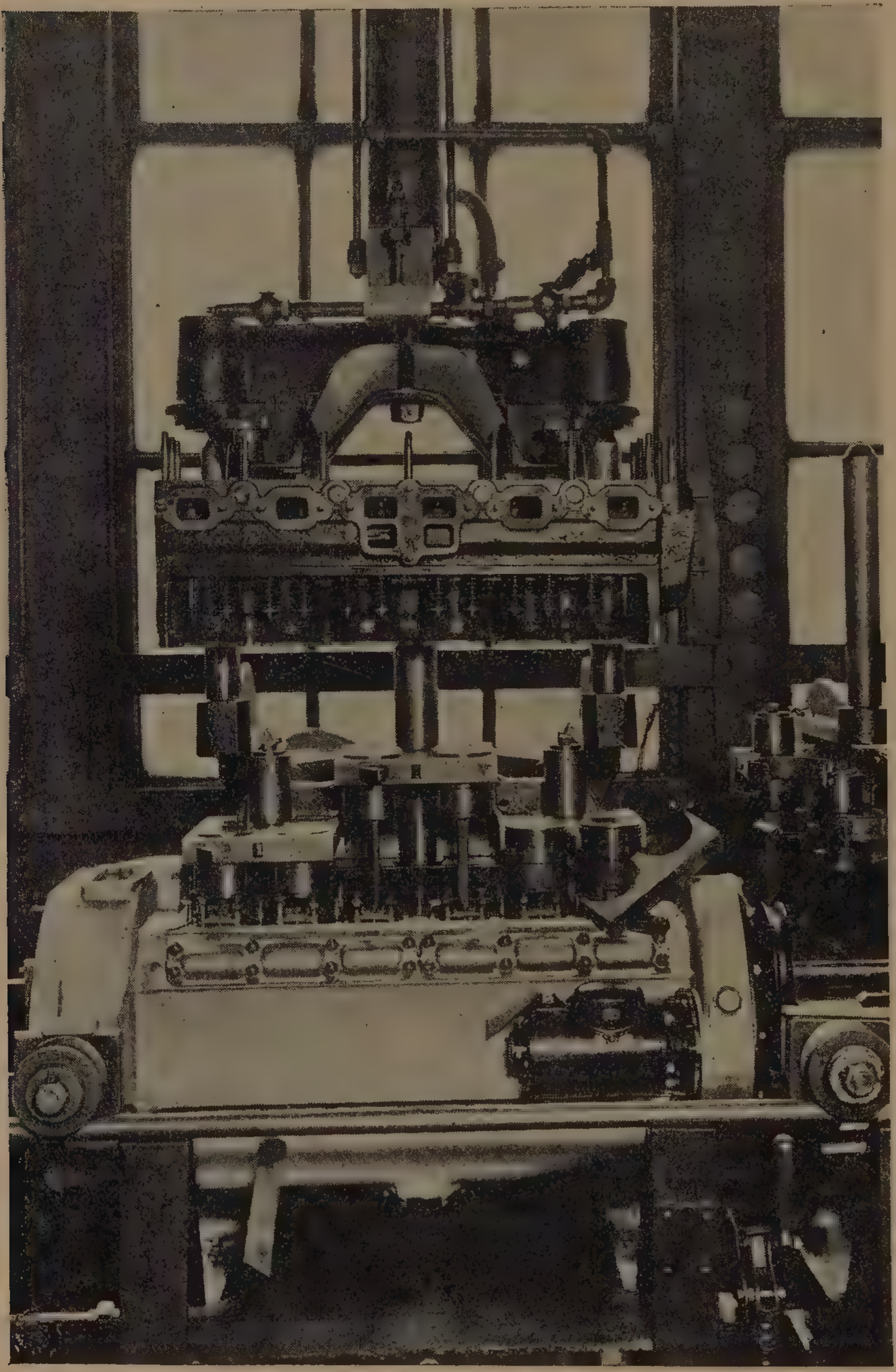


FIG. 414.—FIXTURE FOR PASSING CYLINDER BLOCK OVER PISTONS WITH RINGS (PACKARD).

manifolds, spark plugs, spark plug leads, horn bracket, starter, etc., are put in place, one after another. During these latter operations the engine has been in an upright position, and it is now turned upside down for mounting parts that go inside the crankcase, such as the oil pump, oil leads and oil gauge, as well as the oil pan and cam gear housing cover.

In the case of a unit power plant the clutch and gear case must be attached and the completed power plant is then taken on a conveyor to the testing room.

Piston-Cylinder Assembling Fixture—When the cylinder block and crankcase are separate castings, the assembling operations are naturally somewhat different. It has been customary to first assemble the crankcase with the crankshaft, connecting rods and pistons, and then slip the cylinder block over the pistons while they project from the openings in the top of the crankcase. This is a very delicate operation, particularly in six- and eight-cylinder vertical engines. Unless the block is handled so as to keep it absolutely horizontal, the piston at one end may enter its cylinder first, and the block then constitutes a lever arm by means of which the connecting rod of the piston which has entered may be very easily bent.

A fixture which makes it easy to get all of the pistons with their rings into the cylinders without danger of injury to rods or rings has been developed by the Packard Motor Car Co., and is illustrated in Fig. 414.

A cylinder block is picked up by means of an air hoist and moved across an aisle in the plant directly over the crankcase, which is supported on a wheeled rack on a pair of rails. The arm of the air hoist is provided with a heavy yoke at the bottom, which fits squarely against the top surface of the cylinder block. One stud at each end of the block is a close fit in a hole in the yoke, and hand nuts are screwed on the studs and hold the block to the yoke.

There is also a stationary member to the assembling fixture, which carries an air cylinder whose ram moves a cross-slide. On the cross-slide there are three heads, each for one pair of pistons, and provided with a clamp that squares up and contracts the piston rings. On top of this stationary member there are two pilot arbors. As the ram of the air hoist with the cylinder block is lowered, two long sleeves integral with the yoke pass over the pilot arbors and thus bring the cylinder bores absolutely in line with the pistons.

Operation of Fixture—Before the block is let down the flywheel has been locked against rotation by means of a link and a pin, in such a position that the two center cranks extend straight up. Assuming the crankcase to be in place opposite the fixture, air is admitted to the horizontal air cylinder, which advances the cross-slide into the proper position to hold all pistons centrally over the crankshaft. The three clamps are then applied, contracting the rings and centering the pistons with the center line of the block, which latter is then lowered by means of the hoist, being guided by the vertical arbors. Each of the three clamping heads has a short vertical slide, which allows them to be moved down by the weight of the block until all of the pistons are in the cylinder bores well beyond the lower ring. While the block is being lowered, air is being admitted to a small vibrator or tapper on top of the yoke, which replaces the hand hammering and wrenching usually necessary when lowering the block over the pistons in the ordinary way.

When the block has been lowered to the limit of the slides, the clamps are removed and the cross-slide is withdrawn by reversing the position of its air valve. Coiled springs under the clamp heads automatically return the latter to their topmost position. The cylinder block is then lowered further, until its bottom flange reaches the studs, where a slight tapping may be necessary. The hand nuts holding the block to the yoke are then removed and the ram is raised and moved across the aisle ready to pick up another block.

Running-in by Electric Motor—After being completely assembled the engine is ‘‘run in’’ to limber it up. Owing to the small clearances allowed in the bearings it sometimes takes nearly as much torque to turn the engine over as it later develops under full load. All of the work done in ‘‘running-in’’ is converted into heat, and in order that this heat may be carried off promptly and injury to the bearing surfaces prevented, it is generally well to flood the engine with oil during this operation.

At the Packard plant the running-in of unit power plants is carried on as follows: The engines are run in by electric motors which are connected to the tailshaft of the transmission. For one hour the engine is run in on high gear, that is, at the speed of the electric motor, 625 r.p.m.; then comes a 4-hr. run on second gear, which gives the engine a speed of 1000 r.p.m. During this running-in process the engine is gone over by the tester to locate leaks, scores and any other defects which can be located under the prevailing conditions (with several other engines running in the same room).

When the engines enter the running-in room a dynamometer inspection card is attached to them, on which all defects noted during the running-in and in the dynamometer test are marked down. As a rule, the only entry made on the card during the run-in is the power absorbed by the electric motor, which is a measure of the friction horse power of the engine. Motors of a 15-hp. rating are used.

When the running-in period is over the engines are taken to a bench, where the spark plugs are inserted and the ignition is timed. This timing is effected by means of a mark on the flywheel, the spark being set to occur 1½ in. ahead of the dead center position when the spark lever is fully advanced.

Test in Silent Room—Next the engines are moved along an overhead track to the dynamometer room, in which latter there are fifteen silent rooms, each of which contains a Sprague 150 hp. dynamometer. Each engine remains in one of these rooms for about two hours, the following tests being made: After the engine is fastened down to the stand and all controls are linked up, the generator brushes are inspected, all nuts are inspected for tightness, the line-up of valves and push rods is checked, and the pressure of the carburetor spring is measured and adjusted.

The engine is then started with the dynamometer, the oil level is tried, valves are tried for riding, and any compression leaks are located and stopped. Next the engine is started under its own power, and while thus running the generator is checked for groans and open circuit, and the ammeter reading is verified. Knocks and scores of the cylinders are watched for. The engine is then given a hard run for 10 minutes, during which it is watched carefully, all nuts are tightened and leaks, cracks and sand holes located.

The engine is then throttled to observe its low-speed performance. A power curve is taken over the range 400-1800 r.p.m. The specific fuel consumption is checked by running the engine under a constant dynamometer load until one-twentieth of a gallon is consumed. The oil-pressure relief valve is adjusted and the cylinders are tried with a Locophone for knocks and scores.

After various tests on the clutch and transmission, the tire pump, starter gears and front end gears are tried; the engine is examined for oil, gasoline, exhaust and air leaks, and is then throttled down until it begins to misfire, or at least to 100 r.p.m. The carburetor is then checked, the engine is idled for 5 min., and smoky exhaust is watched for, the push rods are set, and the compression in each cylinder is measured with a standard compression gauge. At the conclusion of this routine test the engines are removed from the dynamometer stand.

CHAPTER XXVI

Engine Tests

In speaking of engine tests, a distinction is made between routine tests and development tests. There is, of course, still another class, viz., that of research tests, the object of which is to determine the characteristics of engines and the interrelations of various operating factors without a view to developing a marketable product. Routine tests were briefly covered in the previous chapter and we will therefore pass to a discussion of development tests, for which a variety of instruments have been evolved. The simplest of these is the pressure gauge, by means of which either the compression or the explosion pressure in any cylinder can be determined. A very practical instrument of this kind is the Okill, which is made in England.

Okill Pressure Gauge—Ordinarily gaseous pressures are measured by the Bourdon tube gauge or steam gauge, but this is applicable only where the pressures are comparatively steady. In a case like that under consideration, where the pressures to be measured are maintained only for an instant, it is very important that the effects of inertia be eliminated, or at least minimized. In the Okill gauge the principle of balancing one pressure against another is made use of, so that when the reading is taken there is no movement of the parts, and it therefore cannot be influenced by inertia effects.

The body of the instrument (Fig. 415) screws into the spark-plug hole or compression-cock hole, and forms a cylinder in which is located a short piston in the form of a cylindrical plug with a flange at its upper end. Against the flanged top of the piston bears one arm of a bell crank—the other arm of which forms an indicating hand—and a hollow piston rod and oil tube. At its lower end the piston rod is provided with a flanged head F, the point of which bears against the piston and the flange of which, by abutting against the lower end of the barrel C, prevents the piston from being blown out of the cylinder when the spring is adjusted while the engine is operating. The head on the piston rod serves as the lower support for the spring.

The spring is compressed by turning on the knurled head of the instrument, with the result that a threaded sleeve to which the head is secured is screwed further into the barrel C. The ends of the spring are anchored to its two supports, so that as the knurled head is turned the flanged head is turned equally, and the motion of the knurled head, which measures the compression of the spring and therefore its pressure, is transmitted through suitable gearing to a counter at the side of the gauge. This counter, which comprises a number of dial wheels, is so designed that it indicates the pressure in pounds per square inch which must be exerted by the gas against the piston of the instrument in order to balance the spring pressure.

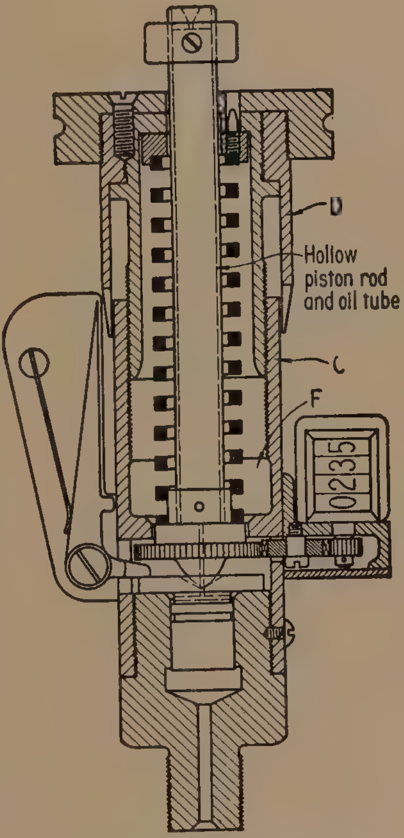


FIG. 415.—OKILL GAUGE.

Operation of Okill Gauge—In operation, the pressure of the spring ordinarily holds the flange of the piston tight against its seat. If the maximum pressure occurring in the engine cylinder into which the instrument is screwed exceeds the pressure of the spring, each impulse will raise the piston, and the succeeding impulses, therefore, will cause the indicating hand to vibrate. By means of the knurled head the threaded sleeve is then screwed down until vibration of the hand cease, and the reading of the counter then shows the cylinder pressure in pounds per square inch. For reading compression pressures the spark is cut off from the cylinder to which the gauge is applied, while the rest of the cylinders operate in the normal way. When determining the explosion pressure the engine is fired on all cylinders.

In some cases where the spark plug hole is at the bottom of a pocket and it is therefore impossible to insert the type of instrument shown, the indicating bell crank and the counter are left off. The scale of gas pressures is then engraved on the shell, and balance of the spring and gas pressures is determined by lightly pressing with the finger against the top of the piston rod.

Figure 416 is a plot of compression and explosion pressures against engine speed, made from data obtained with an Okill indicator on a four-cylinder Vauxhall engine. The short vertical lines show the variations in compression and explosion pressures of the different cylinders. This chart and the drawing of the indicator are reproduced from a paper on Engine Testing Instruments read by J. Okill before the Institution of Automobile Engineers.

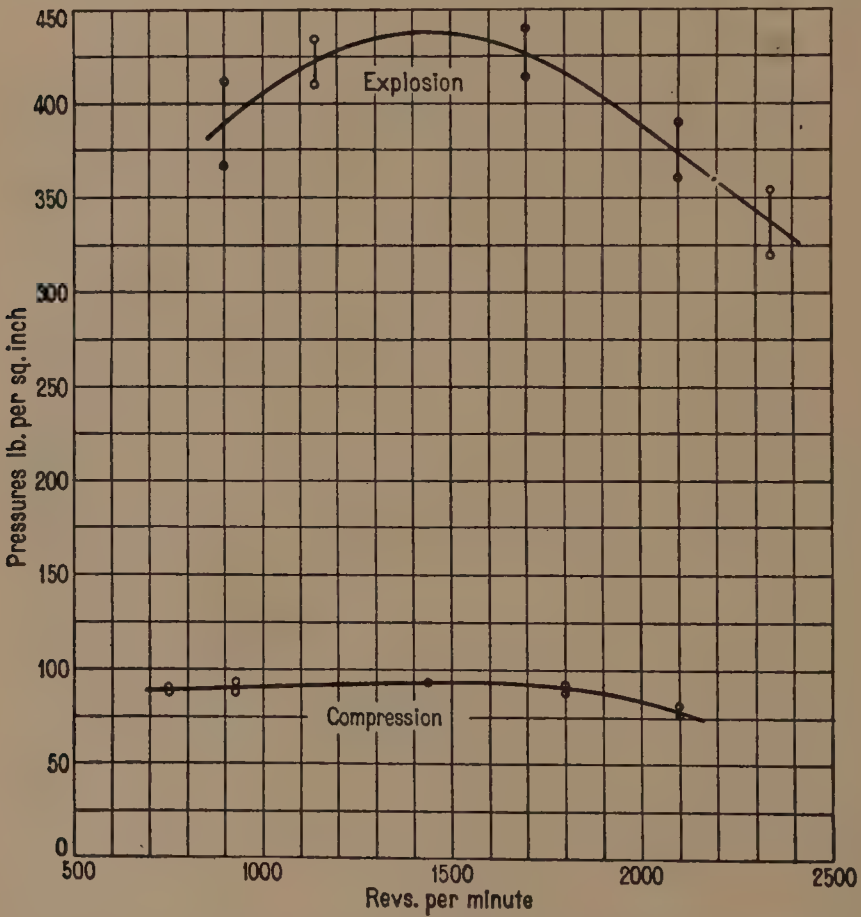


FIG. 416.—COMPRESSION AND EXPLOSION PRESSURES DETERMINED WITH OKILL GAUGE.

Piston Type Indicators—The ordinary piston type indicator as illustrated in diagram in Fig. 6 and briefly described in Chapter III, which is much used on low-speed steam engines, is unsuitable for use on high-speed internal combustion engines because of the inertia of its moving parts, which makes it impossible for them to follow the rapid variations of the cylinder pressure. A method has been devised however by K. J. de Juhasz (of the Technical College of Budapest) whereby the indicator cylinder is placed in communication with the engine cylinder for only a small fraction of each cycle. Hence the pressure on the piston of the instrument is constant and inertia effects are eliminated.

The stroboscopic valve used consists of two slides mounted side by side in a suitable guide, there being a port in each of the slides and in each side of the guide. From the port in one side of the guide connection is made to the cylinder to be indicated, and from the other to the cylinder of the indicator. Communication must be established only once during a revolution of the crank operating the slides, and crank angles and port locations are so determined that this occurs when the slides are moving at their maximum speeds; so that the duration of the opening is as small as possible.

The stroboscopic valve is driven positively from the engine through a phase-shifting mechanism combined with it. Referring to Fig. 417, this mechanism consists of a planetary reduction gear of the internal gear type. Shaft A, which is connected to the engine crankshaft, carries a spur pinion B meshing with two planetary pinions C, capable of revolving on studs in the toothed disc E, which latter is mounted loosely on shaft A. Planetary pinions C also mesh with internal gear F, which has twice as many teeth as pinion B and is keyed to the crank of the stroboscopic valve. Hence this shaft rotates at one-half engine speed and makes one revolution for each engine cycle.

Since the two slides of the valve are connected to cranks almost opposite each other, the ports will register twice during each revolution of the crank, but by making the angle between the two cranks slightly less than 180°, they are made to register once when in line with the ports in the housing and once when out of line with same, in which latter case no communication is established.

Meshing with toothed disc E is a worm G, by means of which the disc can be revolved around its axis and the phase relation of the valve to the engine cycle changed. The toothed disc also meshes with spur gear H, having one-half the number of teeth, and this spur gear, therefore, will revolve at crankshaft speed; hence the motion of the indicator drum can be derived from it. For this purpose a crankpin is secured into the side of gear H, carrying a connecting rod I moving in a slot of the housing of the phase-shifting mechanism.

Optical Indicators—During the early years of the automobile industry two high-speed engine indicators (or manographs) making use of optical levers, known as the Hospitalier and the Schulze, were developed in France and were used for some time in many automobile works. They gave volume-pressure diagrams, the same as the ordinary piston type indicator, but there were so many chances for error in their use that when, during the War, intensive research work on aircraft engines became necessary, several new types of indicator were developed, in which some of the previous faults were eliminated. The high-speed indicator, however, still remains a delicate and somewhat uncertain instrument which is suited only to the laboratory and skilled hands.

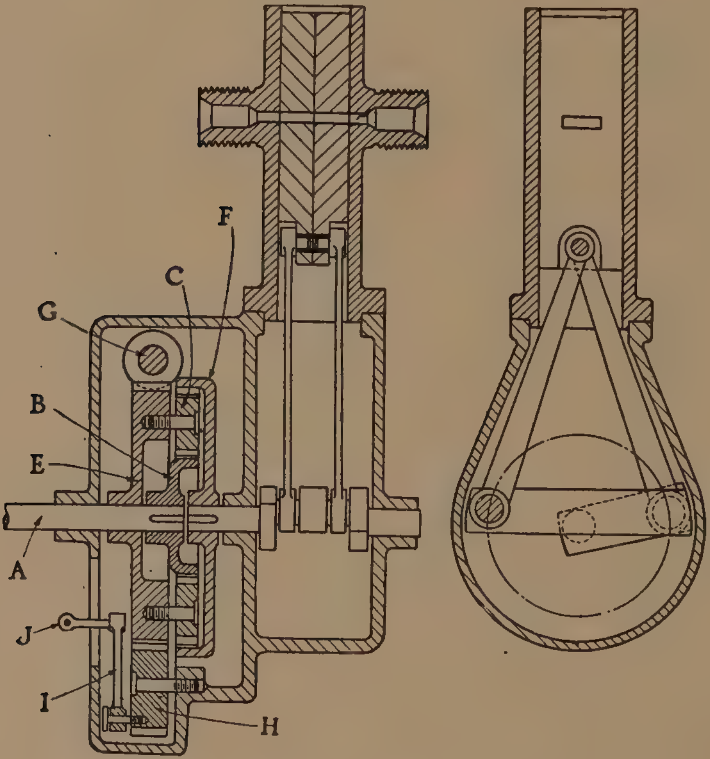


Fig. 417.—Juhasz Stroboscopic Indicator Valve.

Midgley Indicator—An indicator giving either a pressure-time or a pressure-volume record or diagram has been developed by Thomas Midgley, Jr., of the General Motors Corp. It comprises three principal elements, viz., the pressure unit, which contains a small piston on which the gaseous pressure acts; the indicator unit, a box containing a lamp housing, motor and glass frame; and a timing device, consisting of a base with a crankshaft attachment and switches.

In the manographs referred to in the preceding paragraph the cylinder gases acted on a spring steel diaphragm. In order that the temper of the diaphragm might not be drawn, the diaphragm chamber had to be placed at some distance from the engine, and this necessitated a connecting tube of small bore and considerable length, which was one source of error. Midgley obviated this by making use of a piston located inside the tubular pressure element and the head of which is flush with the inner surface of the combustion chamber wall when the pressure element (Fig. 418) is screwed in place. A steel spring of rather unusual form presses against the piston, which is bolted to a central rod extending through the spring element and connecting to a concave mirror held by a horizontal pivoted arm. The spring is of great strength and the motion of the piston under the varying pressure of the gases is exceedingly small.

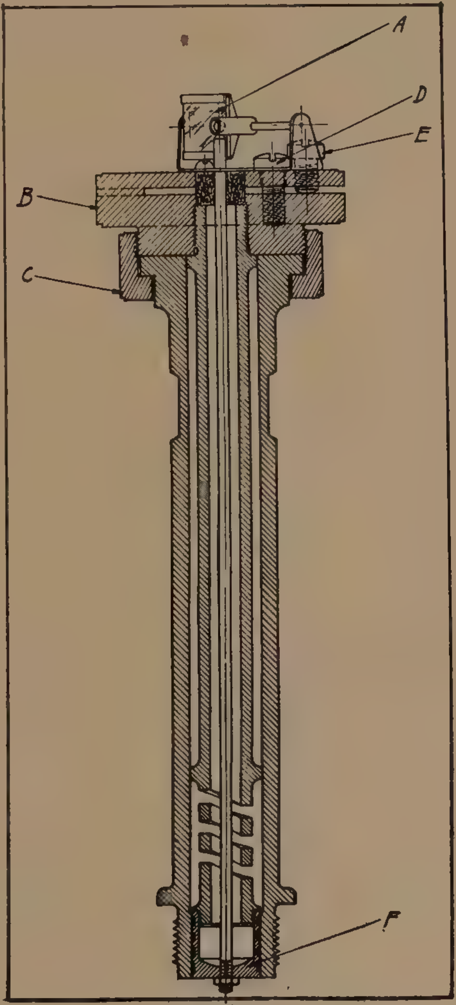


Fig. 418.—Pressure Element of Midgley's Indicator.

A sectional view of the indicator unit is shown in Fig.

419. A beam of light from the filament of a miniature electric lamp A passes through a small hole B in the wall of the housing and strikes the concave mirror of the pressure element by which it is reflected through opening C onto an octagonal mirror D, whence it is reflected onto the ground glass plate E in the housing which forms one-eighth of a cylindrical shell. As the concave mirror of the pressure element is tilted up more or less by the gas pressure, the reflected beam will strike the octagonal mirror at different heights and at different inclinations, and the spot of light on the ground glass E will travel up and down with the point of reflection on the octagonal mirror, only its range of up and down travel will be almost twice as great.

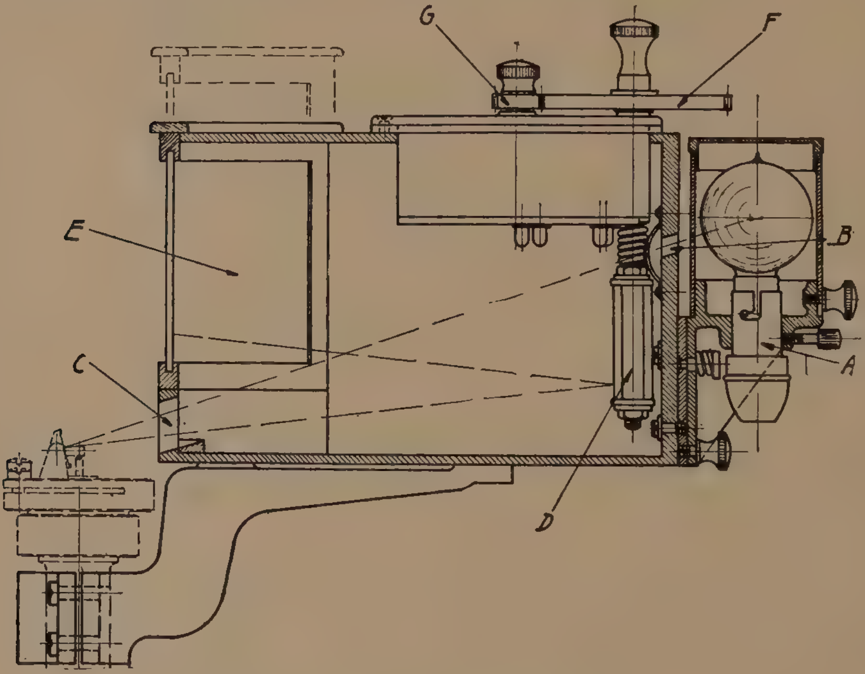


FIG. 419.—MIDGLEY INDICATOR UNIT.

Therefore, if the octagonal mirror were stationary the spot of light on the glass plate would play up and down through a distance proportional to the pressure variations in the engine cylinder. This mirror, however, is adapted to move around its axis, and the character of the motion given to it depends upon whether a pressure-volume or a pressure-time diagram is wanted.

Pressure-Volume Diagrams—For a pressure-volume diagram the mirror is merely rocked slightly back and forth through the timing device which is illustrated in Fig. 420. This comprises a crank of 1½-in. throw, with means for bringing it into phase with the engine crankshaft. The crank drives through a connecting rod of adjustable length to the cross-head shown at the left in Fig. 420. In order that correct diagrams may be shown, the ratio of connecting rod length to crank throw must be the same in this reducing mechanism as in the engine under test, hence this ratio must be variable for the reducing mechanism. From the tip on the cross-head at A a wire is run over a pulley to a lever arm which is put in the place of the gear wheel F in Fig. 419. This will give an oscillating motion to the octagonal mirror so that the spot of light thrown onto the curved glass plate will move horizontally in accordance with the motion of the engine piston and vertically in accordance with the cylinder pressure.

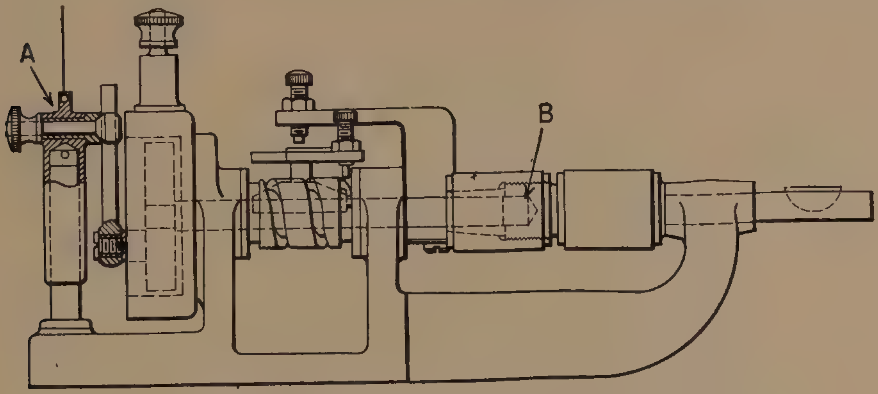


Fig. 420.—Timing Device of Midgley Indicator.

Pressure-Time Diagrams—For obtaining pressure-time cards, the octagonal mirror is rotated in synchronism with the engine crankshaft by means of a synchronous electric motor actuated by current impulses controlled by the timing device. As soon as the spot of light reflected by one face of the octagonal mirror reaches the far end of the ground glass plate, the beam reflected by the following face enters the plate at the near end. The pressure curve corresponding to two strokes extends entirely across the plate.

Newell Point-by-Point Indicator—An indicator by means of which the pressure in the cylinder at any particular point of the cycle can be accurately determined was developed at the Bureau of Standards in Washington in connection with aircraft engine development work and was described in Report No. 107 of the National Advisory Committee for Aeronautics, by H. C. Dickinson and F. B. Newell.

The pressure element of this indicator is shown in section in Fig. 421. A thin metal diaphragm 1 divides the chamber into two parts, the lower one communicating with the combustion chamber through a number of comparatively short passages, as shown. This close connection to the engine has the advantage of obviating pressure lag on the diaphragm, but it necessitates water cooling of the connection, for which purpose the jacket 3 is provided.

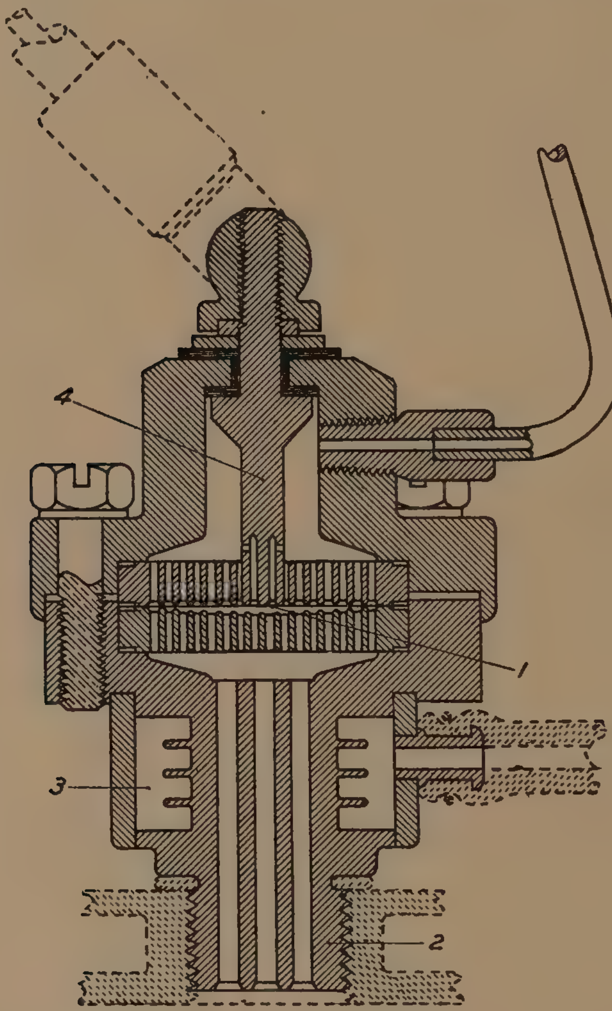


FIG. 421.—PRESSURE ELEMENT OF NEWELL INDICATOR.

A balancing pressure on top of the diaphragm is supplied by compressed air or other gas, which arrives through a fine copper tube and fills the space in the upper portion of the element, whence it passes through the perforations in the support of the disc to the diaphragm chamber.

When the pressures on its opposite sides are out of balance, the diaphragm is forced against one or the other of two circular brass plates, which are about 3/16 in. thick, perforated with No. 60 drill holes, and provided with circular corrugations on which the diaphragm bears, the motion of the latter being thus limited to 0.005 in. The upper supporting surface is plane, while the lower is concaved. These supports prevent distortion of the diaphragm beyond its elastic limit.

When a test is being made, the pressure on the upper side of the diaphragm is substantially constant for minutes at least, while the pressure on the under side passes through a complete cycle during each engine revolution. Consequently, at high engine speed the diaphragm vibrates at a rate of many times per second. In normal operation the pressures on opposite sides of the diaphragm are balanced at only two points in the cycle, and at all other times the diaphragm is pressed against one support or the other, depending upon which of the pressures is greatest.

The diaphragm is a metal disc (usually steel, plated to protect it against corrosion), 1¼ in. in diameter and from 0.003 to 0.006 in. thick. For a source of pressure a bottle of compressed air or liquid CO_2 is used, and for the under pressure to balance the vacuum in the cylinder during the inlet stroke, a water aspirator. Pressures of from 1 to 2 atmospheres absolute are measured by means of a mercury gauge; of from 15 to 100 lbs. per sq. in. gauge, by means of a 100-lb. Bourdon gauge, and above this by means of a 1000-lb. Bourdon gauge.

Electrical Circuits of Indicator—When the diaphragm moves from one support to the other there is equality of pressures, and this motion is recorded by making or breaking an electric circuit. The central part, 4, of the upper support is insulated from the rest of the pressure element, and is included in a circuit containing a telephone receiver serving as a detecting instrument, and a battery, one side of which latter is grounded to the engine frame. When the diaphragm moves upward it closes the circuit, thereby clicking the telephone.

Clicking of the telephone, which occurs normally twice during each cycle, shows that at certain phases of the cycle there is equality of pressures on opposite sides of the diaphragm, but it gives no means of determining at what points of the cycle this occurs. This function is served by the timer (Fig. 422), a rotating member in synchronism with the engine cycle and adjustable in phase with respect to it, which closes the circuit for only about 1° of arc during each cycle. The timer consists of a disc of insulating material, in the circumference of which is inlaid a narrow strip of brass, with which a steel brush is adapted to contact. An auxiliary device on the timer measures the angle at which the ignition spark passes.

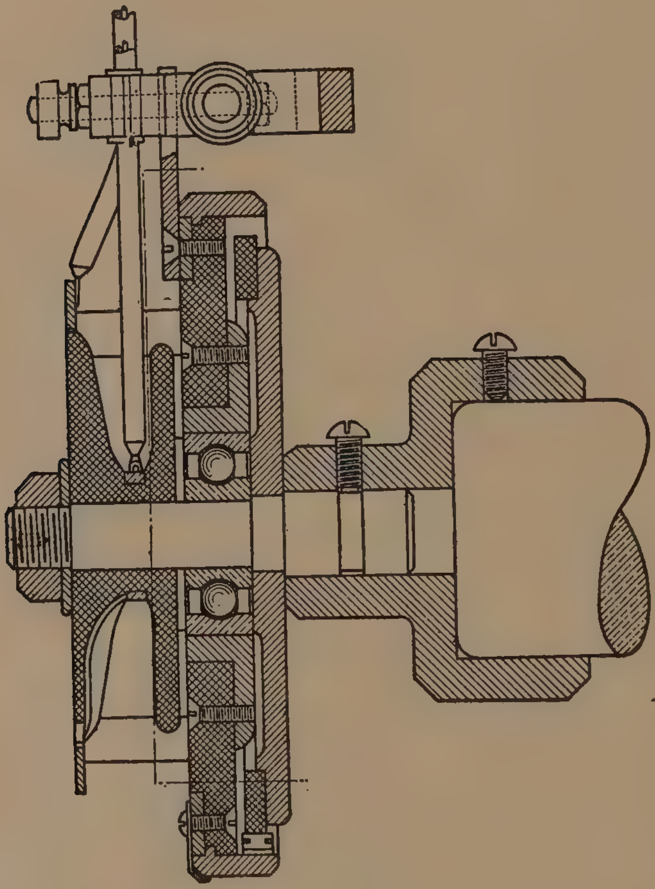


Fig. 422.—Timer of Newell Indicator.

Operation of Newell Indicator—A diagram of the arrangement of the various parts is shown in Fig. 423. Pressure control is effected by means of a number of ⅛-in. needle valves, and the small testing supply tank is filled to any desired pressure from the main tank and then serves as the source of pressure. The water aspirator (shown in diagram) and the cooling jacket of the pressure element are connected

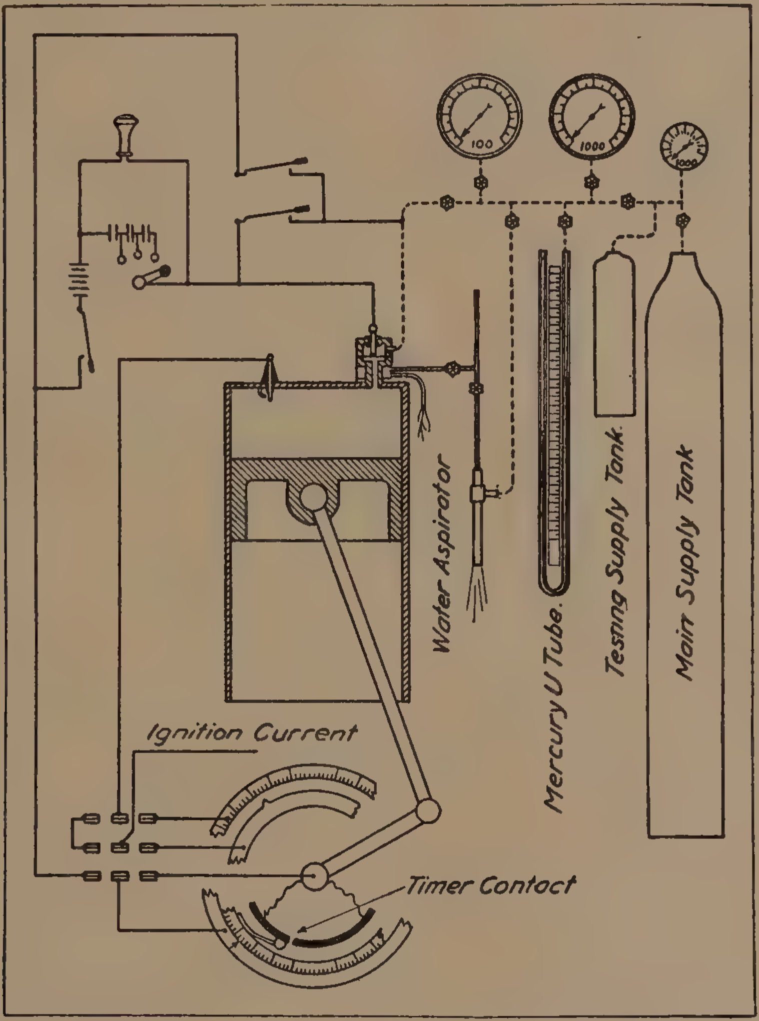


Fig. 423.—Diagrammatic Representation of Newell Indicator Connected to Engine.

in the same circuit. All electrical circuits are shown in diagram on the left side.

At high engine speeds the time of contact at the timer segment is so short that the current through the telephone receiver cannot build up to the proper value to give an audible click. In that case a condenser is shunted across the phone, which will become charged when the circuit is closed and then discharges through the phone, thus intensifying the click. At low speeds the click is sometimes disagreeably loud, and it can then be dulled by connecting a larger condenser in paralled with the phone.

In use, the sensitivity and zero reading of the instrument must be checked occasionally, by leaving one side of the diaphragm chamber open to the atmosphere and measuring the pressure required on the other side to obtain a balance, that is, to close or open the circuit. If the zero reading is large or the sensitivity poor, the diaphragm should be replaced. The timer is set so it registers zero angle when the engine crank is in the dead center position.

This instrument can be used for determining the maximum compression pressure or the maximum explosion pressure, in which case the timer is short-circuited and the applied pressure is gradually increased. Clicks in the telephone will then occur in pairs, following each other closely and separated by longer intervals. The two clicks of a pair will draw closer and closer together as the applied pressure approaches the pressure of compression or explosion, and when this pressure is reached the clicks will disappear.

For measuring the pressures at other points of the cycle the timer must be in circuit. In the indicator head the circuit is then clôsed as long as the engine pressure is higher than the applied pressure, but the timer closes the phone circuit for only 1° of arc. Thus the timer contact serves to determine whether the circuit is open or closed at a selected point of the cycle. If a click occurs in the telephone, it shows that at that particular point of the cycle the engine pressure is higher than the applied pressure, and vice versa. The point of balance is therefore marked by cessation of the clicking. Observations are preferably taken by two operators, as the measurements are then completed in a much shorter time and there is less chance for conditions to vary materially.

Methods of Plotting Diagrams—The test results consist in a set of pressure figures in lbs. per sq. in. and corresponding crank angle figures, and the results can then be plotted in a number of ways. One way is to plot them in the form of a polar diagram. There will be two curves on this diagram, one showing the variations of pressure in the cylinder during the compression and expansion strokes; the other (which is preferably plotted to a larger scale) the pressures during the exhaust and inlet strokes. A second plan consists in plotting a pressure-time curve, in which case crank angles are plotted along the horizontal and pressures along the vertical axis;

a third plan is to plot the conventional pressure-volume diagram, with piston displacements along the horizontal and pressures along the vertical axis. Finally, a fourth method consists in plotting piston displacement on the horizontal axis and pressures on the vertical axis, both according to logarithmic scales. An advantage of the latter method is that it magnifies the ordinates of the suction and exhaust curves, thus making it possible to determine the values represented by these curves with a higher degree of accuracy.

Farnboro Indicator—An indicator based to a certain extent on the same principles as the Newell, but tracing a pressure-time card, was developed by Major Norman and H. Wood of the Royal Aircraft Establishment (England) and is being manufactured by Dobbie-McInnes of Glasgow. It is known as the Farnboro indicator. The pressure unit contains a light disc of stainless steel having an insulated guide and closing an electrical circuit each time it ends up against either supporting surface, that is, each time there is equality of pressure. It has a range of motion of 0.010 in. and weighs only 0.003 lb.

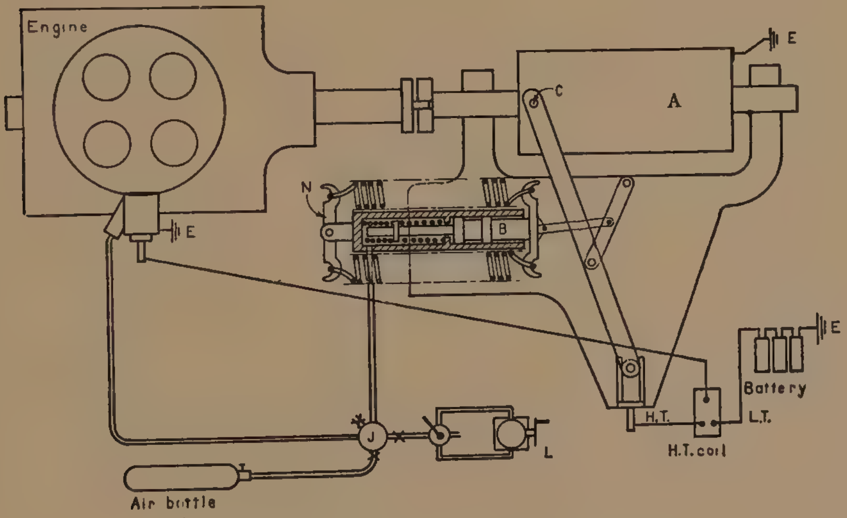


FIG. 424.—FARNBORO RECORDING INDICATOR.

The set-up of the Farnboro indicator is shown in Fig. 424. At the left may be seen a single cylinder engine with four valves in the head. The pressure element is secured into the side of the cylinder, and its insulated terminal is grounded at E. From the air bottle a tube extends to the valve J, whence other tubes lead to the pressure element at the cylinder, to a pressure cylinder containing a piston B and to a hand pump L which by means of a change-over valve can be made to act either as a suction or pressure pump and serves to give the balancing pressures for the inlet and exhaust strokes.

Connected to the piston B by a suitable linkage is a lever arm which is electrically insulated and has fitted at its free end a spark point C moving over the drum A. This drum is rotated either at camshaft speed or at crankshaft speed. In the first case the pressure record of a complete cycle is spread over the circumference of the cylinder; in the second, it extends twice around the cylinder.

It was stated in the foregoing that each time the pressures on opposite sides of the pressure disc are balanced, an electrical contact is broken and remade. This contact device is included in the primary circuit of a spark coil indicated in Fig. 424, in the lower right-hand corner. The resulting high tension impulse in the secondary coil causes a spark to jump from spark point C to the metal drum. Current for the spark coil is supplied by a three-cell battery, of which one terminal is grounded. By placing a sheet of black paper around the cylinder, the spark will puncture it and thus burn a record on it. As long as the cyclic pressures and also the balancing pressure remain the same, the spark will always pierce the paper in the same place. If now the balancing pressure is increased, the spark point will move farther to the right on the cylinder, and as this higher extraneous pressure will be balanced by the cylinder pressure at a different point of the cycle, the new puncture will be displaced both axially and circumferentially, relative to the previous one. Thus by changing the balancing pressure from one limit to the other of the cyclic pressure within the engine, and back again, a complete pressure-time diagram for the whole cycle is traced. To be able to properly analyze this diagram it is, of course, necessary that the dead center lines be located. This is done by turning the engine over slowly by hand to the end of the compression stroke. Then a spark can be made to pass to the drum by breaking the primary circuit by hand, and the point thus marked on the aluminum drum can be later transferred to the paper, and a line scribed on it parallel to the drum axis.

Phase Adjustment—With optical indicators the first operation in making a test, after the instrument has been set up and connected to the engine, consists in adjusting for phase equality. If the engine is turned over by hand during the compression stroke the pressure in the cylinder will rise to near the normal compression, and during the following down stroke it will drop practically along the same line. If there were no leakage and no interchange of heat between the gases and the cylinder walls the compression and expansion curves would absolutely coincide. As a matter of fact, supposing the cylinders to be tolerably airtight and to be cold, the compression and expansion curves when the instrument is properly synchronized will be substantially as in Fig. 425.

Phase adjustment can also be made with the engine running but with the spark fully retarded. Then, if the phase adjustment is correct, the indicator will retrace the upper end of the compression curve, before ignition occurs. If the indicator leads the engine the return line will be slightly above the top end of the compression line, whereas if it lags, the return line will be below the top end of the compression line and will cross it when ignition occurs.

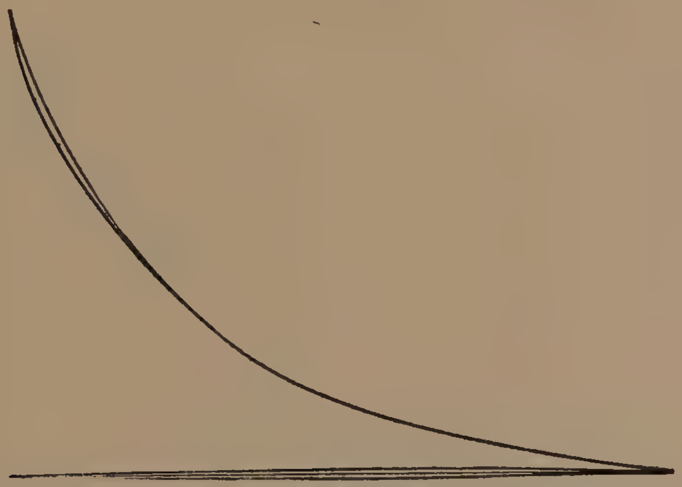


FIG. 425.—COMPRESSION DIAGRAM USED FOR PHASE ADJUSTMENT.

Studying Ignition and Combustion—In some investigations, however, it is advantageous to have the instrument out of phase with the engine, as, for instance, when trying to determine the relative rates of combustion of different fuel mixtures. This factor is indicated by the inclination of the explosion line in the diagram. When the instrument is adjusted in phase with the engine the explosion takes place while the piston is stationary or nearly so, and any slight variation in the rapidity of combustion, therefore, does not show distinctly on the diagram. On the other hand, if the instrument is adjusted so as to be 90° out of phase with the engine, the piston motion, and, consequently, the horizontal motion of the beam of light, will be a maximum while the explosion occurs, and any variation in the rate of explosion is shown on a much larger scale. A diagram taken with the instrument out of phase by about 90° is shown in Fig. 426.

Use of Indicator—One use of the indicator consists in determining the indicated horse power. By subtracting the brake horse power from the indicated horse power, the internal friction losses and the mechanical efficiency of the engine are found.

The area enclosed by the compression and expansion curves represents work done upon the piston by the expanding gases.

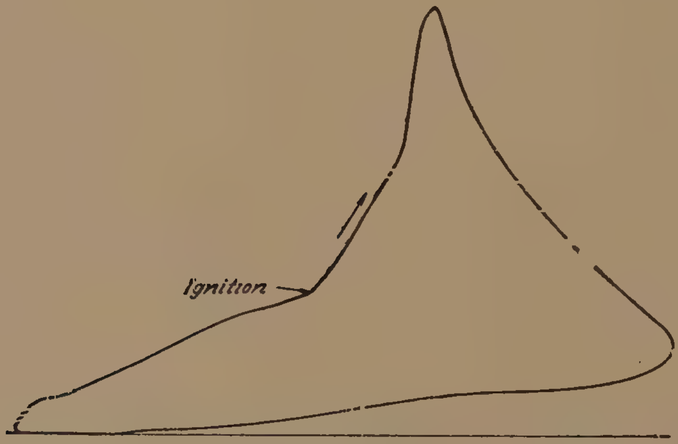


Fig. 426.—Diagram Taken with Crank Motion 90° Out of Phase.

The exhaust, compression and suction curves form a loop and the area enclosed by them (the shaded area in Fig. 427) represents negative work and must be subtracted from the positive work represented by the area enclosed by part of the exhaust and compression curves and the expansion curve. In order to find the indicated horse power it is necessary to determine the mean height of the diagram. After the mean height of the positive work area and the mean height of the negative work area have been determined and their difference has been taken, the mean effective pressure is found by multiplying this difference by the scale of the diaphragm, which is generally expressed in pounds per square inch per inch. The power developed by the cylinder from which the diagram was taken may then be determined from the formula

$$\text{I.H.P.} = \frac{M.E.P. \times \pi b^2 \times l \times n}{4 \times 2 \times 12 \times 33{,}000} = \frac{\pi b^2 l \times M.E.P. \times n}{3{,}168{,}000},$$

where *M.E.P.* is the mean effective pressure as determined from the diagram; b the bore of the cylinder in inches; l the length of stroke in inches and n the number of revolutions per minute. The speed of the engine must be taken at the same time as the diagram, if the indicated horse power is to be determined.

The other and, perhaps, the most important use of the indicator consists in determining faults in the operation of the engine due to defective valve setting, ignition, carburetion, etc. If the engine works under the proper conditions it gives a diagram very closely approximating the ideal diagram, as discussed in Chapter III. From the known data of the cylinder under test such a diagram may be drawn, and the diagram obtained by means of the instrument may then be compared with it. A good diagram taken with an optical indicator is shown in Fig. 428. Diagrams indicating too early and too late ignition are shown in Fig. 429.

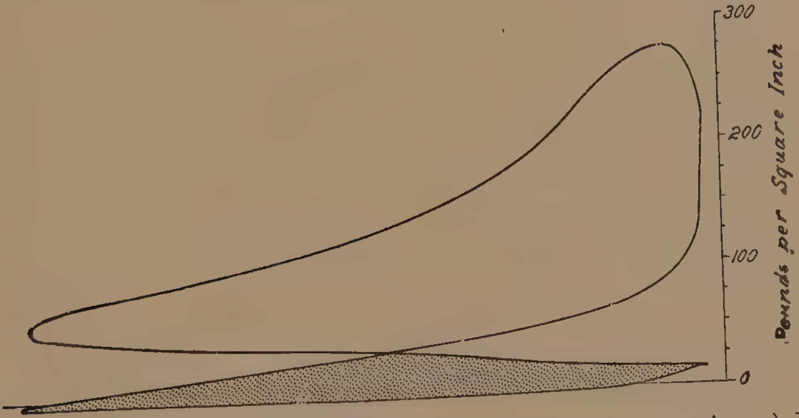


FIG. 427.—DIAGRAM SHOWING NEGATIVE WORK (SHADED AREA) DUE TO HIGH EXHAUST BACK PRESSURE.

Pressure-Time Diagram—Many modern indicators trace pressure-time diagrams. These have the advantage over pressure-volume diagrams that the phenomena occurring near the dead center positions are shown much more clearly with respect to their time relations. On the other hand, the pressure-time diagram is not so familiar to engineers as the

FIG. 428.—TYPICAL ENGINE INDICATOR DIAGRAM.

pressure-volume diagram, and, besides, it does not permit of determining the mean effective pressure directly.

Pressure-time diagrams can be readily converted into pressure-volume diagrams if blank cards on transparent paper are made available, on which vertical lines are ruled corresponding to the piston positions corresponding to crank positions at 10° intervals throughout the stroke. It is then only necessary to transfer from the pressure-time diagram the pressure ordinates corresponding to these various crank positions to the

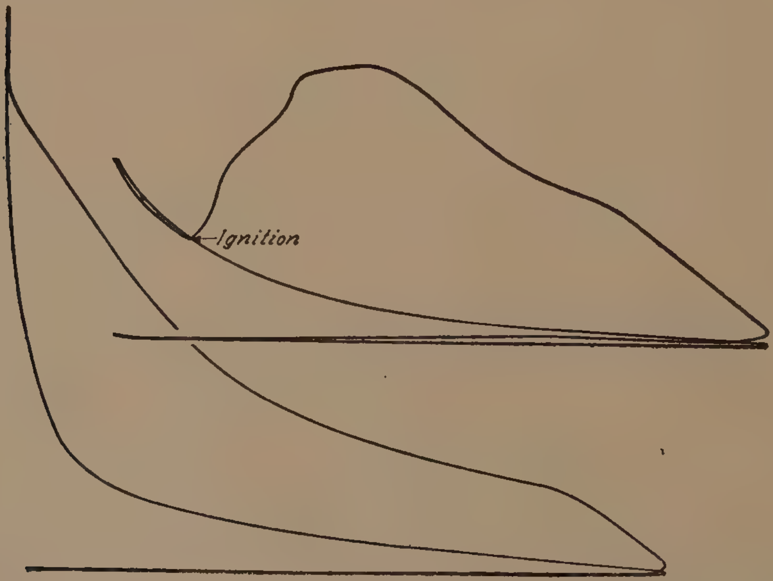


FIG. 429.—DIAGRAMS SHOWING TOO EARLY AND TOO LATE IGNITION.

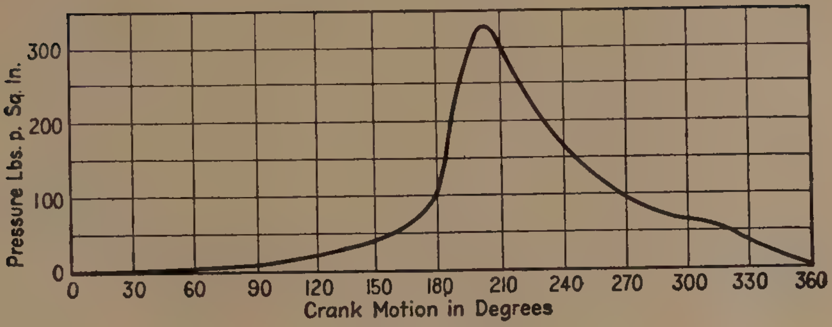


FIG. 430.—PRESSURE-TIME DIAGRAM.

pressure-volume card by shifting the card, and then connecting the points. After some experience this can be done in a few minutes. A typical pressure-time diagram is shown in Fig. 430.

Logarithmic Diagram—It was suggested by Clayton in the Journal of the American Society of Mechanical Engineers in 1912 that the pressures and volumes be plotted on loga-

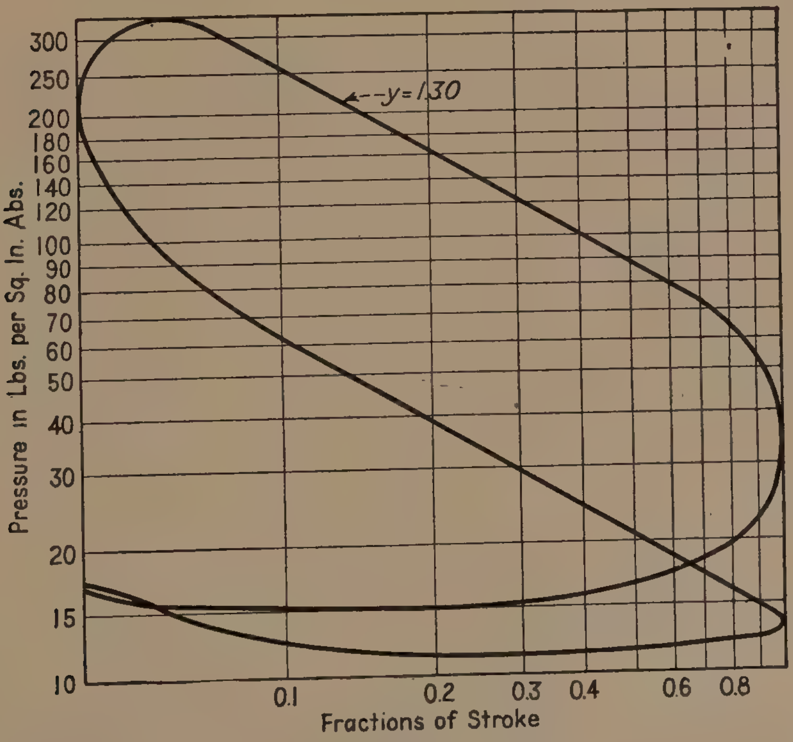


FIG. 431.—LOGARITHMIC PRESSURE-VOLUME DIAGRAM (COMPRESSION AND EXPANSION LINES STRAIGHT).

rithmic scales. A diagram of this type, relating to test results from a foreign six-cylinder automobile engine at 2000 r.p.m., is shown in Fig. 431. This engine gives its maximum output at about 3000 r.p.m.

Such a diagram has two advantages. It permits of drawing the exhaust and intake lines on the same sheet as the compression and expansion lines, using the same scale and still having variations of pressure during these two strokes clearly shown; and it makes the compression and expansion lines straight, provided they obey the law pv^{γ} = constant. The inclinations of these lines, moreover, are a measure of the values of γ for compresison and expansion, respectively.

Dynamometers—The indicator is of use chiefly in determining improper gas flow, which, as a rule, is due to defects in design. It is mainly a laboratory instrument, and is used only on engines while being developed experimentally. In connection with such problems as the best valve timing, sizes of valves and sizes and location of ports in two-stroke engines, the indicator may render valuable service. However, the chief aim in engine development is always increased brake horse power, and for this reason the testing device most generally used is some form of power brake or dynamometer which enables the tester to determine the power output directly.

An apparatus for determining the power output of an engine is known as a dynamometer. Dynamometers are divided into two classes, viz., absorption dynamometers, which absorb and dissipate the energy output of the engine, and transmission dynamometers, which are inserted between the engine and its load and do not absorb any part of the output. All of the dynamometers which are practically employed for testing automobile engines are of the first kind, i.e., absorption dynamometers.

The Prony Brake—The most primitive form of dynamometer is known as the Prony brake. It consists of a pair of blocks which can be applied to the rim of the flywheel or of a pulley secured to the flywheel or crankshaft, with adjustable pressure. The friction between the blocks and the pulley will absorb the output of the engine and convert it into heat. The blocks are secured to an arm whose other end rests upon the platform of a beam scale or is attached to a spring scale. Referring to Fig. 432, when the engine rotates in the direction indicated by the arrow, it will cause the scale to indicate a pressure. This pressure is due partly to the friction of the blocks on the drum and partly to the weight of the arm. Since it is desired to determine the friction alone, either a correction must be made for the weight of the arm resting on the scale or else the effect must be eliminated by providing a counterweight on the side opposite the arm, so the brake balances around the center of the bore of the brake blocks.

Let the radius of the drum be r and the distance of a vertical line through the point of contact on the scale, from the center of rotation, D. Then, when the engine is running and the scale indicates a pressure W, the friction at the circumference of the drum is evidently DW/r. If the speed of revolution of the engine be n, then the rubbing speed at the rim of the pulley or flywheel is $2\pi rn$ in. per min., and the power dissipated in friction is

$$\text{H.P.} = \frac{D}{r}W \times \frac{2\pi rn}{12 \times 33{,}000} = \frac{DWn}{63{,}025}.$$

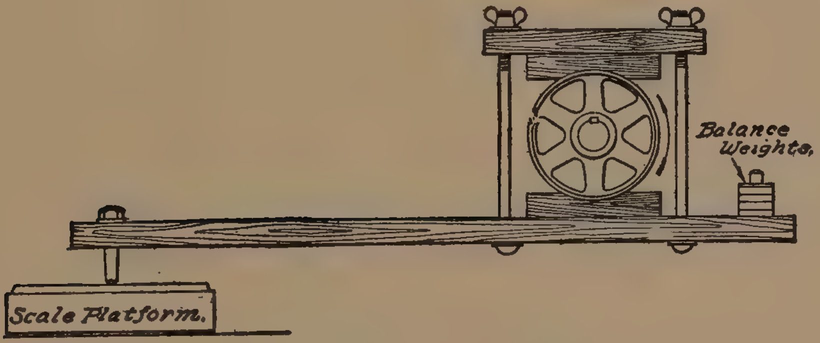


FIG. 432.—PRONY BRAKE.

The Prony brake is not a very practical instrument and is not used for testing automobile engines. It is referred to here only as an historical type and as an aid in explaining the principles of the absorption dynamometer. Some difficulty is experienced in carrying off the heat generated quickly enough. The friction at the circumference changes almost continually, owing to wear and to the expansion due to heating, and for this reason it is not possible to extend Prony brake tests over considerable periods under fixed conditions.

The Fan Dynamometer—The fan dynamometer, which was first introduced in 1902 by Colonel Renard, of the firm of Panhard & Levassor, of France, consists merely of a wooden beam bolted to the end of a horizontal shaft and carrying two square discs of aluminum, which may be moved farther in toward the shaft or farther out from it, as desired. Small sizes of the dynamometer may be mounted directly upon the crankshaft of the engine to be tested, but in the larger sizes it is preferable to fix the beam to a special shaft which is mounted in ball bearings in a bearing housing bolted to a supporting frame. At its free end this shaft is provided with a member of a universal joint, so it can be connected to the crankshaft by means of an intermediate shaft and two universal joints.

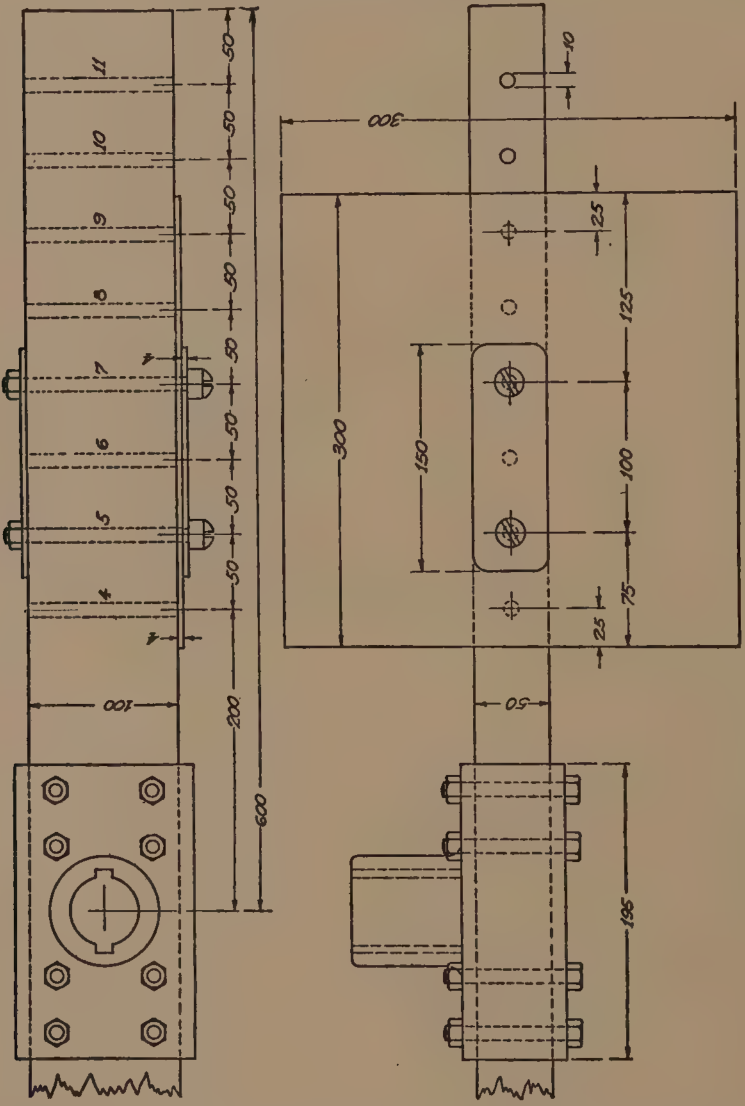


FIG. 433.—DIMENSION DRAWING OF FAN DYNAMOMETER.

As the dynamometer was originally constructed by Colonel Renard the beam was made of ash and provided with two aluminum plates secured by two bolts each. To allow of the distance of the plates from the center line of rotation being varied, each arm of the beam was drilled with eleven bolt holes, equidistant from each other. These eleven holes admitted of ten different positions of the plates. Ten intermediate positions were obtainable by means of extra holes in the plates, as shown in Fig. 433. The distance between holes was 5.5 centimeters (2.2 in.), the total length of the instrument 52.8 in., and the complete weight 26.4 lbs.

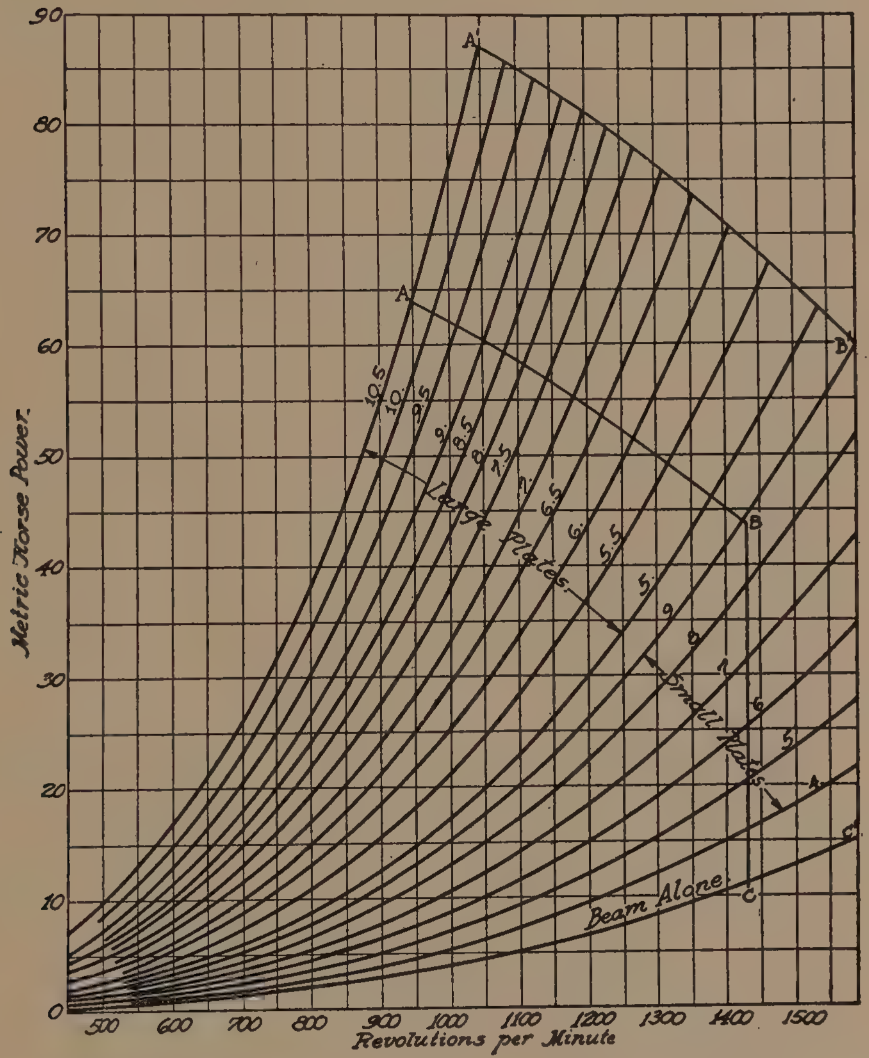


FIG. 434.—CALIBRATION CURVES OF FAN DYNAMOMETER.

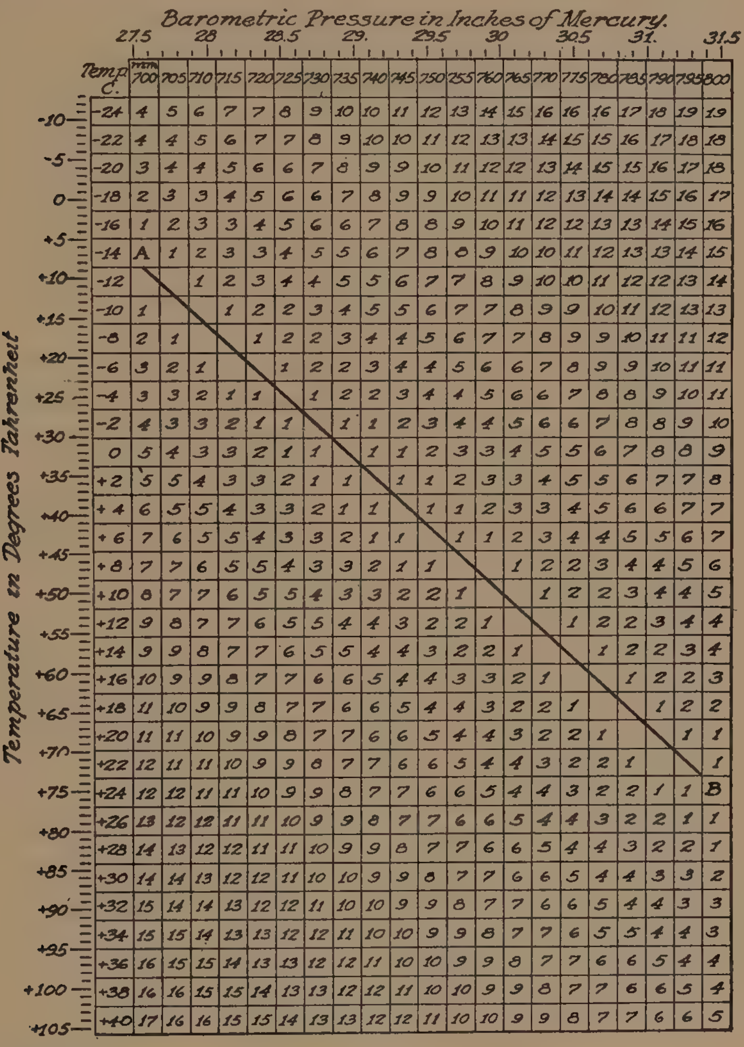
The resisting torque of a fan dynamometer is proportional to the square of the angular velocity, and the power absorbed to the cube of the angular velocity. Both the resisting torque and the power absorbed are also proportional to the density of the atmosphere, which varies with the thermometer and barometer readings.

It has been proved both by calculation and by experiment that for fans of proportional dimensions throughout the power coefficient varies as the fifth power of the ratio of linear dimensions. This law was verified by means of experiments with two instruments the linear dimensions of which bore to each other the ratio of 2.2. The fifth power of 2.2 being 51.54, the coefficient of the larger dynamometer should be equal to that of the smaller one multiplied by 51.54. In spite of the inevitable experimental errors and inaccuracies of construction, this relation was found to hold very well for all positions of the plates on the beam.

When a dynamometer is constructed with dimensions of revolving parts bearing to each other relations different from those given in the table below, it is necessary to calibrate the instrument by means of some standard form of dynamometer. Then, when it is desired to build larger or smaller instruments and adhere to the same proportions of parts, all that is necessary is to multiply the power factor found for the first instrument by the fifth power of the ratio of the linear dimensions in the original and new instruments respectively. Colonel Renard found a difference of less than one-half of 1 per cent in the power indications of a number of dynamometers of the same size constructed with ordinary care, and he expressed a belief that with all-metal construction the difference would be smaller. The relative dimensions of the parts of the Renard fan dynamometer, expressed as a function of the modulus a (distance between adjacent bolt holes), are as follows:

Distance between holes	a
Length of beam	24 a
Thickness of beam	a
Width of beam	2 a
Side of square discs	60/11 a
Thickness of discs	4/55 a
Diameter of bolts	10/55 a
Diameter of shaft	40/55 a
Number of holes in each arm of beam	11
Distance from the center to the outermost hole	11 a

A calibration chart for a fan dynamometer of the Renard design is given in Fig. 434. The calibration figures apply under normal atmospheric conditions, that is, 29.92 in. of mercury and 60° F. temperature, under which atmospheric air weighs 0.0765 lb. per cu. ft. The weight of air for any



Temp. C.	mm 700	705	710	715	720	725	730	735	740	745	750	755	760	765	770	775	780	785	790	795	800
-24	4	5	6	7	7	8	9	10	10	11	12	13	14	15	16	16	16	17	18	19	19
-22	4	4	5	6	7	7	8	9	10	10	11	12	13	13	14	15	15	16	17	18	18
-20	3	4	4	5	6	6	7	8	9	9	10	11	12	12	13	14	15	15	16	17	18
-18	2	3	3	4	5	6	6	7	8	9	9	10	11	11	12	13	14	14	15	16	17
-16	1	2	3	3	4	5	6	6	7	8	8	9	10	11	12	12	13	13	14	15	16
-14	A	1	2	3	3	4	5	5	6	7	8	8	9	10	10	11	12	13	13	14	15
-12			1	2	3	4	4	5	5	6	7	7	8	9	10	10	11	12	12	13	14
-10	1			1	2	2	3	4	5	5	6	7	7	8	9	9	10	11	12	13	13
-8	2	1			1	2	2	3	4	4	5	6	7	7	8	9	9	10	11	11	12
-6	3	2	1			1	2	2	3	4	4	5	6	6	7	8	9	9	10	11	11
-4	3	3	2	1	1		1	2	2	3	4	4	5	6	6	7	8	8	9	10	11
-2	4	3	3	2	1	1		1	1	2	3	4	4	5	6	6	7	8	8	9	10
0	5	4	3	3	2	1	1		1	1	2	3	3	4	5	5	6	7	8	8	9
+2	5	5	4	3	3	2	1	1		1	1	2	3	3	4	5	5	6	7	7	8
+4	6	5	5	4	3	3	2	1	1		1	1	2	3	3	4	5	6	6	7	7
+6	7	6	5	5	4	3	3	2	1	1		1	1	2	3	4	4	5	5	6	7
+8	7	7	6	5	5	4	3	3	2	1	1			1	2	2	3	4	4	5	6
+10	8	7	7	6	5	5	4	3	3	2	2	1			1	2	2	3	4	4	5
+12	9	8	7	7	6	5	5	4	4	3	2	2	1			1	2	2	3	4	4
+14	9	9	8	7	7	6	5	5	4	4	3	2	2	1			1	2	2	3	4
+16	10	9	9	8	7	7	6	6	5	4	4	3	3	2	1			1	2	2	3
+18	11	10	9	9	8	7	7	6	6	5	4	4	3	2	2	1			1	2	2
+20	11	11	10	9	9	8	7	7	6	6	5	4	4	3	2	2	1			1	1
+22	12	11	11	10	9	9	8	7	7	6	6	5	4	4	3	2	2	1			1
+24	12	12	11	11	10	9	9	8	7	7	6	6	5	4	4	3	2	2	1	1	B
+26	13	12	12	11	11	10	9	9	8	7	7	6	6	5	4	4	3	2	2	1	1
+28	14	13	12	12	11	11	10	9	9	8	7	7	6	6	5	4	4	3	2	2	1
+30	14	14	13	12	12	11	10	10	9	9	8	7	7	6	6	5	4	4	3	3	2
+32	15	14	14	13	12	12	11	10	10	9	9	8	7	7	6	6	5	4	4	3	3
+34	15	15	14	13	13	12	12	11	10	10	9	9	8	7	7	6	5	5	4	4	3
+36	16	15	15	14	13	13	12	12	11	10	10	9	9	8	7	7	6	6	5	4	4
+38	16	16	15	15	14	13	13	12	12	11	10	10	9	9	8	7	7	6	6	5	4
+40	17	16	16	15	15	14	13	13	12	12	11	10	10	9	9	8	7	7	6	6	5

FIG. 434-A.—CORRECTION-FACTOR CHART FOR USE WITH CALIBRATION CURVES.

other pressure and temperature can be calculated from the equation

$$W = \frac{1.3256P}{459.6 + T} \text{ lb. per cu. ft.,}$$

developed in Chapter II, and a suitable correction applied, the power absorbed varying directly with the density of the air.

Fan dynamometers are of service for routine tests, where a large number of engines of the same type must be tested quickly. For research and development work they are not suited, because it is impossible to vary the torque independently of the speed without stopping the engine. The degree of accuracy attainable is also rather low, the results being influenced by the nearness of walls, by air drafts, etc. In Europe a type has been built which is provided with an adjustable housing, which to a certain extent overcomes the objection of lack of controllability. For the sake of safety the fan must be surrounded by a substantial cage. The attractive features of the fan brake are its simplicity and low cost, but its disadvantages seem to outweigh these and it has practically passed out of use.

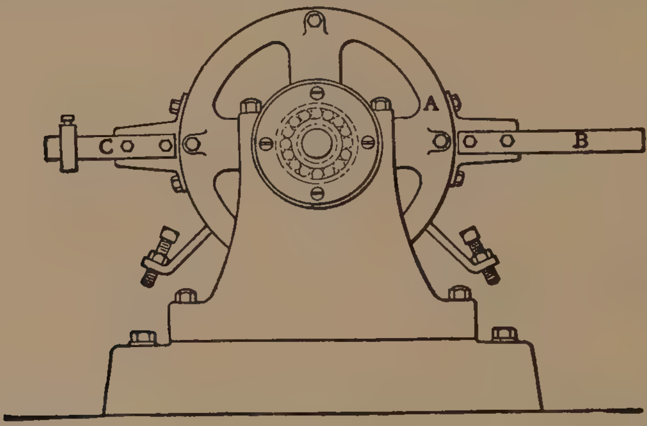


Fig. 435.—Diagram of Electric Cradle Dynamometer.

Electric Cradle Dynamometer—The most practical and convenient device for making accurate horse power tests on an engine is the electric cradle dynamometer. This consists of a dynamo of the annular field frame type. The field frame, instead of being rigidly bolted to the base or cast integral therewith, is supported in two large radial ball bearings on pedestals rising from a cast iron base. The armature shaft is connected to the shaft of the engine to be tested through a flexible coupling. A sketch of such a dynamometer is shown in Fig. 435. When the armature is rotated by the gasoline engine it exerts a turning effort on the field frame A and tends to carry the latter around with it. Two radial arms extend from the field frame, one, B, for measuring the pull exerted by means of a scale, and the other, C, for balancing the former. Current is induced in the armature of the generator, the same as in the ordinary dynamo, and this may be used in lighting up a bank of incandescent lamps or it may be dissipated in a water rheostat. The pressure exerted by the lever arm on the scale is equivalent not only to the electromagnetic reaction between the armature and field frame, but also to the armature bearing friction and the commutator brush friction, and the only item not included in this pressure is the friction in the ball bearings supporting the field frame, which is negligible, however. The method used in connection with such a dynamometer for calculating the horse power is exactly the same as that used in connection with the friction brake, and the same formula can be used, i.e.,

$$\text{H.P.} = \frac{DWn}{63{,}025}.$$

As in the cases of all other dynamometers, torque and speed measurements must be made simultaneously, and for this reason a tachometer is generally mounted on top of the dynamometer.

Sprague Electric Dynamometer—Practically all of the electric cradle dynamometers now in service in this country are of the Sprague type, manufactured by the General Electric Co. These are made in numerous sizes from a nominal capacity of 1 hp. (at 1000 r.p.m.) up. Field excitation of these dynamometers is by line current at either 110 or 220 volts, wherever possible, as this permits load control within much wider limits than if self-excitation were used.

In the Sprague dynamometer, at the end of the torque arm there is a short cross-bar, from one end of which depends a link to a balance lever carried on a pivot stand rising from the base. The other arm of this balance lever connects through a link to a spring balance, which hangs from the beam of a beam scale. Thus double means of measuring the torque are provided. The beam scale is convenient when it is desired to develop a given torque, as the proper weights can then be put on and the load adjusted until the scale balances. All joints of the linkage are knife edges.

For convenience in making horse power calculations the dynamometers are designed with an equivalent torque arm length of 63.025 in. The horse power formula given in the foregoing then reduces to

$$\text{H.P.} = \frac{Wn}{1000}.$$

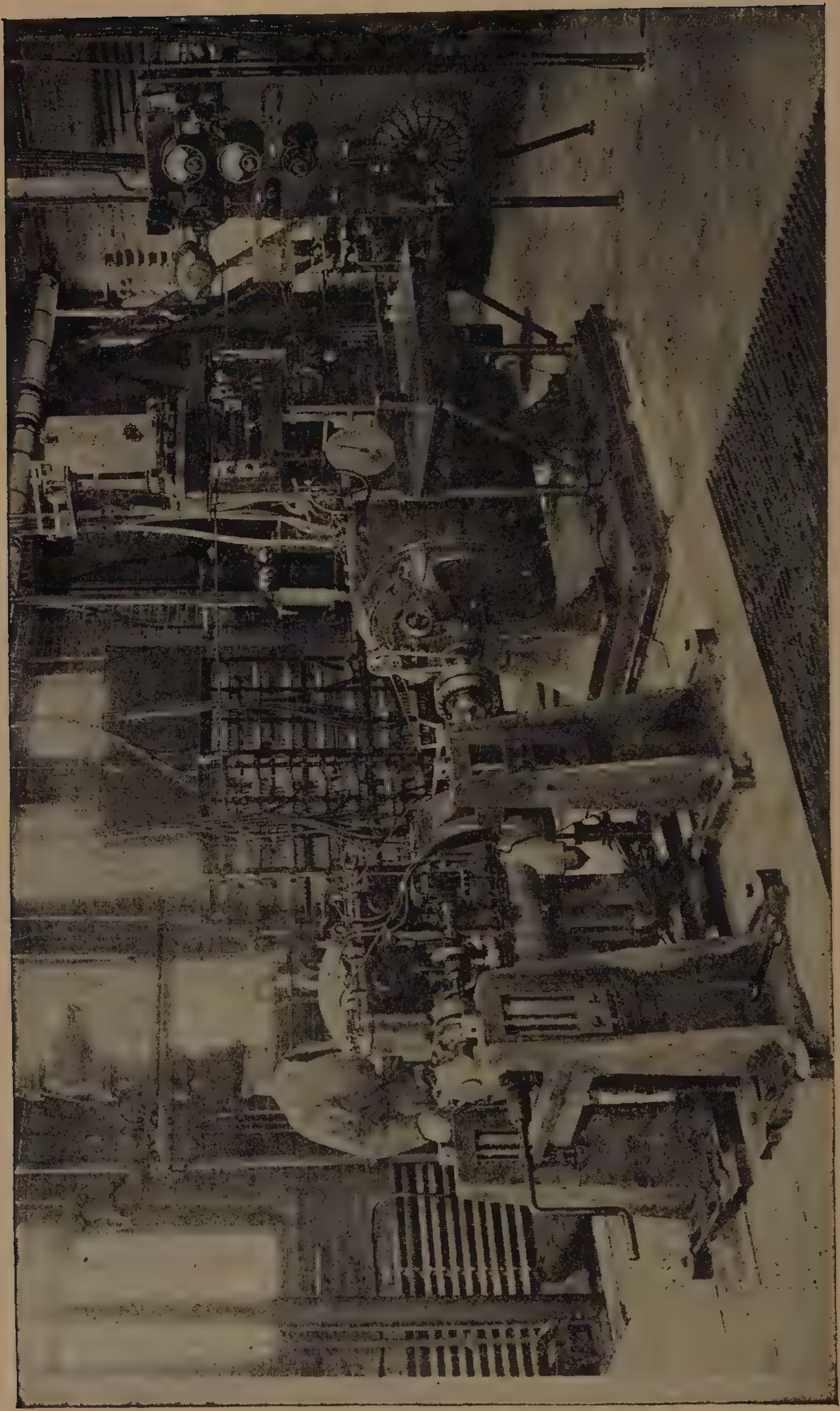


Fig. 436.—Sprague Electric Dynamometer Installed in Testing Laboratory of Iowa State College, Ames, Ia.

A Sprague dynamometer installed in the engine laboratory of Iowa State College, Ames, Iowa, is shown in Fig. 436. A part of such an installation is a control board. This carries at the bottom a dial type rheostat control switch for the main circuit, and above this a dual control rheostat for the field, by means of which very fine adjustments of speed and load are possible. Other items of equipment on the control board include an ammeter, a voltmeter, and an overload circuit breaker with auxiliary ignition switch.

The dynamometer is used also as a starter for starting the engine, and the dial switch at the bottom of the control board by which the electrical load is varied serves also as a starting switch. Friction tests of the engine are made by "motoring" it by means of the dynamometer, and the readings of the voltmeter and ammeter, together with an efficiency curve furnished with the dynamometer, then permit of the power absorbed by the engine while being "motored" being accurately determined.

When the engine is run under load the electrical energy delivered by the dynamometer is absorbed by the resistances shown in the background of Fig. 436.

Water Dynamometers—A type of hydraulic dynamometer used to a limited extent in engine testing is shown in Fig. 437. It comprises a stator and a rotor. The rotor consists of four steel discs, 40 in. in diameter and 1/4 in. thick, securely keyed to a substantial shaft driven from the engine under test through a universal jointed shaft and multiple disc clutch. The brake discs revolve in four separate compartments within the casing, but clear the walls in all directions by 3/8 in., the only metallic contact between the rotor and the stator being in the bearings which carry the shaft. The casing is free to revolve through a small angle and its motion is practically frictionless, owing to the special method of mounting it. An arm projects from the periphery of the casing, the end of which is linked to a steel beam supported on knife edges by a pillar in the same manner as the beams of platform scales, the whole forming a compound lever which is balanced by means of an adjustable weight.

When it is desired to make a brake reading, the rotor is first engaged by means of the multiple disc clutch. Water is then admitted to one of the end compartments near the shaft, and its flow regulated by an ordinary globe valve. Centrifugal force carries the water out to the periphery of the disc, and creates a high pressure, the water then serving as a friction medium between the steel disc and the walls of the compartment. As a result, the casing tends to rotate with the disc

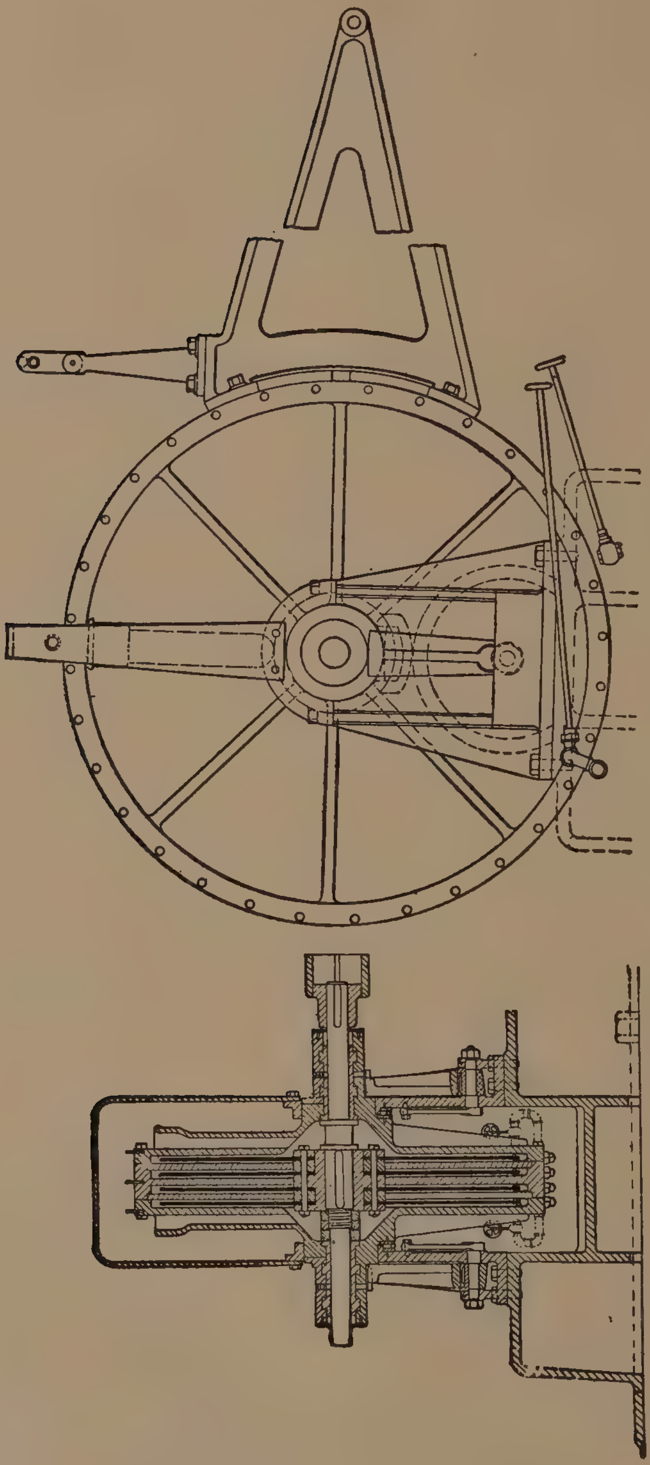


Fig. 437.—Alden Hydraulic Dynamometer.

in direct proportion to the friction, and hence to the torque. Whatever friction results from the rotation of the shaft in the bearings of the brake casing has a tendency to revolve it in the same direction; therefore, the entire power developed by the engine is accounted for by the weighing apparatus, which counteracts the tendency of the casing to rotate.

If one disc or compartment is not capable of absorbing the total power of the engine, the compartment at the other end of the casing can be brought into service, and the water in these compartments may be transferred to the inner compartments by means of by-pass valves, thus bringing all four discs into use in absorbing the maximum power of which the brake is capable. By varying the quantity of water in the compartments a very flexible control is obtained. This is done by regulation of the inlet and outlet, or drain, valves. The mechanical energy of the motor is converted into heat inside the brake and is carried off by the water, the temperature of which is usually kept below 150° F. The brake water, as well as the jacket water, can be cooled in a radiator and used over and over.

Silence Rooms—In order to better be able to detect the slightest unnatural noises in an engine after it has been tuned up, some factories have been provided with "silence rooms" in which the engines are run under dynamometer load after they have been run in the regular testing department. In the latter, where a great many engines are constantly under test, there is naturally a good deal of noise, and observations as to the quiet running of a particular engine are rendered difficult. Figure 438 is a view in such a silence room. These rooms are built with 24-in. brick walls to exclude all outside noises and are equipped with a dynamometer stand. When the engine is mounted on the stand the one door into the room is closed and a test made to determine whether any part gives out any untoward noises, which would indicate faulty machine work or assembling.

Cold Rooms—A good deal of trouble is often experienced with engines in cold weather. In extreme winter weather starting may be almost impossible either because the starter equipment is of inadequate capacity or because no ignitable mixture is formed. To be able to study "cold starting" problems at leisure at any time of the year, some manufacturers have built cold rooms, in which the temperature can be reduced to as low as — 10° F. by suitable refrigerating equipment. The cold room of the Dodge Brothers plant was described in detail in a paper read before the Society of Automotive Engineers at Detroit in Jan., 1926.

Tests of High-Speed Mechanisms—With the increase in engine speeds certain troubles have been run into which it has been difficult to locate and eliminate because the speed of the parts involved makes it impossible to follow them with the naked eye. A considerable variety of instruments has been developed during the past decade which makes it possible for an investigator to clearly observe the relative positions of the parts of such mechanism during every phase of their cyclical motion.

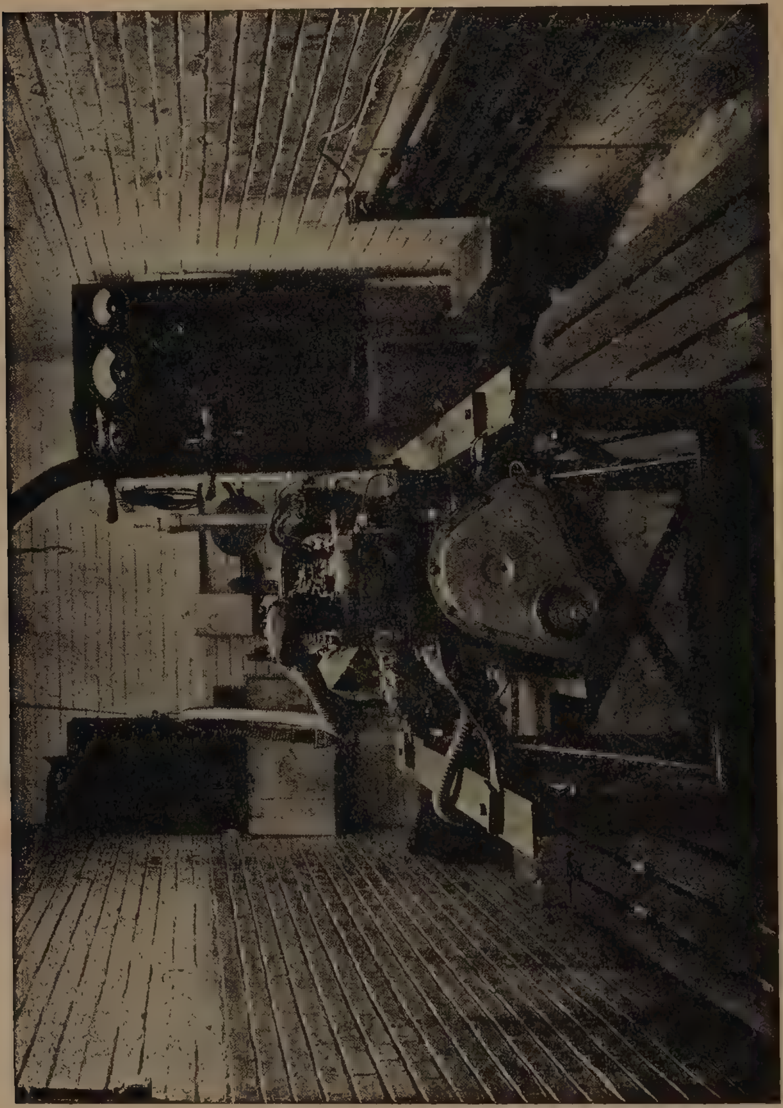


FIG. 438.—SILENCE ROOM FOR DETERMINING UNUSUAL ENGINE NOISES.

The type of instrument probably most familiar is the slow motion moving picture machine. By this, photographic views are taken of the mechanism at the rate of 100 or more a second, and they are then projected on the screen at a much slower rate. A second type of instrument is based on the stroboscopic principle. By means of a system of revolving shutters, operated in synchronism with the part to be observed, the latter is exposed to the eye of the observer for an instant during each cycle, and always at the same part of the cycle. Owing to the persistence of vision, the parts of the mechanism appear stationary to the observer, in the relative positions they occupy while at this point of the cycle. A phase-shifting device is always provided with the stroboscope, and by operating this device the observer can reproduce the actual cyclic motion at a very slow rate.

A third type of instrument depends upon the principle of instantaneous illumination of the mechanism once during every cycle, either by a spark from an induction coil or else by a flash from a vacuum tube (neon lamp). The make-and-break device or commutator must be operated in synchronism with the mechanism to be observed. In the past one difficulty with the last mentioned type of instrument has been inadequate illumination. This difficulty is said to have been eliminated by a recent French invention.

Location of Knocks—Among the more serious defects which may be found in new engines are knocks and other objectionable noises, either in a bearing or elsewhere. When such a defect is present it is readily detected by the ear, but its exact location is a problem that is not so easily solved. Knocking is generally due to looseness in some part, most frequently in the connecting rod bearings, between the cylinder wall and piston, or at the joint of the flywheel with the crankshaft. It requires a free play of only a few thousandths of an inch to produce an audible knock, and any slight defect in the workmanship, therefore, may bring on this trouble. The location of the knock is greatly facilitated by the use of an instrument based on the same principle as the stethoscope, used by physicians in detecting irregularities in the actions of certain organs of the human body.

An instrument of this type made in England is known as the Tectoscope. It comprises a metal rod which is held against the engine or other part, near the supposed source of the noise. The rod fastens into a sheet metal case inside of which there is a diaphragm, this assembly being known as the reproducer. The reproducer connects by a thick-walled rubber tube to a Y-fitting, which latter in turn connects by two similar rubber tubes to a head frame with hard rubber ear tips that are pressed into the ears. A companion instrument, known as a Tectophone, has a horn or mouthpiece secured to the center of the reproducer housing, instead of a metal rod, and is therefore actuated by sound waves in the air, instead of by high frequency vibrations of the metallic parts of the machine tested.

APPENDIX

POWERING OF MOTOR VEHICLES.

When a motor vehicle travels on a level course, the power required to keep it in uniform motion is made up of two items. The first of these is that required to overcome frictional resistances of various kinds, such as friction in the transmission mechanism and wheel bearings, internal friction in the tires, resistance of the road surface to deformation, etc. These various resistances are substantially independent of the speed of the vehicle, but their sum varies widely with the character of the road surface. On the very finest smooth, hard roads these resistances, which together are known as the traction resistance, amount to about 15 lbs. per 1000 lbs. of vehicle weight (including load).

On paved roads in only fairly good or average condition, and when other factors, such as the character of the tires fitted and the degree of inflation, are not the most favorable, the traction resistance is about 25 lbs. per 1000 lbs. of total weight moved.

The other item is the air resistance which is proportional to the projected frontal area of the car and to the square of the speed. It also varies with the form of the car, being a minimum for shapes approaching stream-line form. For an average car the air resistance is given approximately by the equation.

$$R_a = 0.003AV^2 \text{ lb.},$$

where A is the frontal projected area in square feet and V the car speed in miles per hour.

When the car is ascending a grade there is still another resistance to be overcome, which is equal to the product to the total weight moved by the percentage of the grade. Thus the total resistance to vehicular motion at any speed V is

$$R = 0.025W + 0.003AV^2 + gW/100 \text{ lbs.}$$

If V is the speed in miles per hour then $88V$ is the speed in feet per minute, and the work done per minute is

$$W = 88V(0.025W + 0.003AV^2 + gW/100) \text{ ft.-lbs. per min.},$$

which is equivalent to

$$\frac{88V}{33{,}000}(0.025W + 0.003AV^2 + gW/100) \text{ hp.}$$

$$= \frac{2VW}{30{,}000} + \frac{AV^3}{125{,}000} + \frac{gWV}{37{,}500} \text{ hp.}$$

This equation enables us to calculate approximately the horse power required by a car of given weight and projected frontal area to attain a certain maximum speed on good level roads and on grades of any steepness.

The engine power required by various vehicles depends largely upon the gear reduction and the diameter of the driving wheels. Let r be the gear ratio in high speed or direct drive; D, the piston displacement of the engine; W, the weight of the car with normal load, and d, the wheel diameter in inches; the piston displacement factor

$$F = \frac{3825rD}{Wd} \text{ cu. in. per ton-foot.}$$

This equation applies to vehicles of all kinds. It is found that for passenger vehicles F varies between 24 and 45, the average for all American cars being about 32. A large displacement factor gives a snappy car; a small one a sluggish but more economical car.

In the case of motor trucks the displacement factor, based on the weight with full load, decreases as the load capacity increases. It averages 24.5 for 1-ton trucks; 22.3 for 2-ton trucks; 21 for 3-ton trucks and 19.4 for 5-ton trucks. The minimum for all trucks investigated was 13.5 and the maximum 28.5.

Drill Sizes, Cross Sectional Areas and Tensile Strengths of S. A. E. Standard Screws.

		Areas.		Tensile Strengths.		
Nominal Diameter.	Drill Size.	Full Bolt.	Bottom of Thread.	At 20,000 Lbs. Per Sq. Inch.	At 25,000 Lbs. Per Sq. Inch.	At 30,000 Lbs. Per Sq. Inch.
1/4	7/32	0.0491	0.0325	651	814	977
5/16	17/64	0.0767	0.0525	1,050	1,312	1,575
3/8	21/64	0.1104	0.0808	1,617	2,021	2,426
7/16	3/8	0.1503	0.1132	2,264	2,830	3,396
1/2	7/16	0.1963	0.1486	2,972	3,726	4,459
9/16	1/2	0.2485	0.1888	3,777	4,722	5,666
5/8	9/16	0.3068	0.2400	4,800	6,000	7,200
11/16	39/64	0.3712	0.2888	5,776	7,220	8,664
3/4	43/64	0.4418	0.3514	7,028	8,785	10,452
7/8	25/32	0.6013	0.4816	9,633	12,141	14,449
1	29/32	0.7854	0.6463	12,926	16,158	19,389
1 1/8	1 1/64	0.994	0.799	15,980	19,975	23,970
1 1/4	1 9/64	1.2272	1.009	20,180	25,225	30,270
1 3/8	1 17/64	1.4849	1.243	24,860	31,075	37,290
1 1/2	1 25/64	1.7671	1.503	30,060	37,575	45,090

Four Cylinder Piston Displacement.*

Bore,	Stroke, Inches—									
	3¾	3⅞	4	4⅛	4¼	4⅜	4½	4⅝	4¾	4⅞
3	106.0	109.5	113.0	116.6	120.1	123.7	127.2	130.7	134.3	137.8
3⅛	115.0	118.8	122.7	126.5	130.3	134.2	138.0	141.8	145.7	149.5
3¼	124.4	128.5	132.7	136.8	141.0	145.1	149.3	153.4	157.6	161.7
3⅜	134.1	138.6	143.1	147.6	152.0	156.5	161.0	165.5	169.9	174.4
3½		149.1	153.9	158.7	163.5	168.3	173.1	177.9	182.8	187.6
3⅝			165.1	170.3	175.4	180.6	185.7	190.9	196.1	201.2
3¾				182.2	187.7	193.2	198.8	204.3	209.8	215.3
3⅞					200.4	206.3	212.2	218.1	224.0	229.9
4						219.9	226.1	232.4	238.7	245.0

Bore,	Stroke, Inches—									
	4½	4⅝	4¾	4⅞	5	5⅛	5¼	5⅜	5½	5⅝
4⅛	240.5	247.2	253.9	260.5	267.2	273.9	280.6	287.3	294.0	300.6
4¼		262.4	269.5	276.6	283.7	290.8	297.9	304.9	312.0	319.1
4⅜			285.6	293.1	300.6	308.1	315.6	323.2	330.7	338.2
4½				310.1	318.0	326.0	333.9	341.9	349.8	357.8
4⅝					336.0	344.4	352.8	361.2	369.6	378.0
4¾						363.2	372.1	381.0	389.8	398.7
4⅞							391.9	401.2	410.6	419.9
5								422.1	431.9	441.7

Bore,	Stroke, Inches—									
	5	5⅛	5¼	5⅜	5½	5⅝	5¾	5⅞	6	6⅛
3	141.3	144.9								
3⅛	153.3	157.2	161.0							
3¼	165.9	170.0	174.2	178.3						
3⅜	178.9	183.3	187.8	192.3	196.8					
3½	192.4	197.2	202.0	206.8	211.6	216.4				
3⅝	206.4	211.6	216.7	221.9	227.1	232.2	237.3			
3¾	220.9	226.4	231.9	237.4	242.9	248.5	254.0	259.5		
3⅞	235.8	241.7	247.6	253.5	259.4	265.3	271.2	277.1	283.0	
4	251.3	257.6	263.8	270.1	276.4	282.7	289.0	295.3	301.5	307.8

Bore,	Stroke, Inches—									
	5¾	5⅞	6	6⅛	6¼	6⅜	6½	6⅝	6¾	6⅞
4⅛	307.3	314.0	320.7	327.4	334.8					
4¼	326.2	333.3	340.4	347.5	354.6	361.7				
4⅜	345.7	353.2	360.7	368.3	375.8	383.3	390.8			
4½	365.7	373.7	381.6	389.6	397.6	405.5	413.5	421.4		
4⅝	386.4	394.8	403.2	411.6	420.0	428.4	436.8	445.2	453.6	
4¾	407.5	416.4	425.3	434.1	443.0	451.8	460.7	469.6	478.4	487.3
4⅞	429.2	438.6	447.9	457.2	466.6	475.9	485.2	494.6	503.9	513.3
5	451.6	461.4	471.2	481.0	490.8	500.6	510.5	520.3	530.1	539.9

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*Note: The piston displacements given on this page and on the following pages are in all cases below the true displacements, on account of the limitation to one decimal point. For instance, the four cylinder 3 x 3¾ inch engine has a displacement of more than 106.0 cubic inches but, less than 106.1 cubic inches.

Six Cylinder Piston Displacement.

Bore,	Stroke, Inches—									
	3⅝	3¾	3⅞	4	4⅛	4¼	4⅜	4½	4⅝	4¾
3	153.7	159.0	164.3	169.6	174.9	180.2	185.5	190.8	196.1	201.4
3⅛	166.8	172.5	178.3	184.0	189.8	195.5	201.3	207.0	212.8	218.5
3¼	180.4	186.6	192.8	199.0	205.3	211.5	217.7	223.9	230.2	236.4
3⅜		201.2	207.9	214.7	221.4	228.1	234.8	241.5	248.2	254.9
3½			223.6	230.9	238.1	245.3	252.5	259.7	266.9	274.2
3⅝				247.7	255.4	263.1	270.9	278.6	286.4	294.1
3¾					273.3	281.6	289.9	298.2	306.5	314.7
3⅞						300.7	309.5	318.4	327.2	336.1
4							329.8	339.2	348.7	358.1
4⅛								360.8	370.8	380.8
4¼									393.6	404.3
4⅜										428.4
4½										

Bore,	Stroke, Inches—									
	4⅞	5	5⅛	5¼	5⅜	5½	5⅝	5¾	5⅞	6
3	206.7	212.0								
3⅛	224.3	230.0	235.8							
3¼	242.6	248.8	255.0	261.3						
3⅜	261.6	268.3	275.0	281.8	288.5					
3½	281.4	288.6	295.8	303.0	310.2	317.4				
3⅝	301.8	309.6	317.3	325.1	332.8	340.5	348.3			
3¾	323.0	331.3	339.6	347.9	356.2	364.4	372.7	381.0		
3⅞	344.9	353.7	362.6	371.4	380.3	389.1	398.0	406.8	415.7	
4	367.5	376.9	386.4	395.8	405.2	414.6	424.0	433.5	442.9	452.3
4⅛	390.8	400.9	410.9	420.9	430.9	441.0	451.0	461.0	471.0	481.1
4¼	414.9	425.5	436.2	446.8	457.4	468.1	478.7	489.4	500.0	510.6
4⅜	439.7	450.9	462.2	473.5	484.8	496.0	507.3	518.6	529.9	541.1
4½	465.1	477.1	489.0	500.9	512.9	524.8	536.7	548.6	560.6	572.5

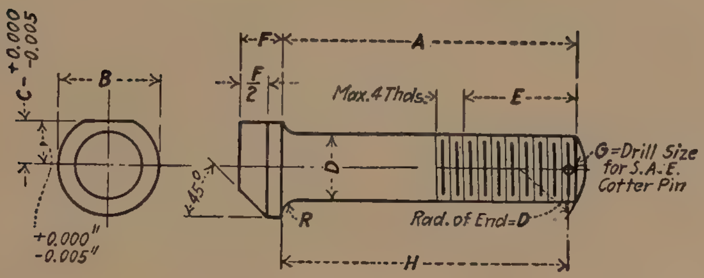
Eight Cylinder Piston Displacement.

Bore,	Stroke, Inches—									
	4	4 1/16	4⅛	4 3/16	4¼	4 5/16	4⅜	4 7/16	4½	4 9/16
2¾	190.0	193.0	196.0	198.9	201.9	204.9	207.8	210.8	213.8	216.7
2 13/16	198.8	201.9	205.0	208.1	211.2	214.3	217.4	220.5	223.6	226.7
2⅞	207.7	210.9	214.2	217.4	220.7	223.9	227.2	230.4	233.7	236.9
2 15/16	216.8	220.2	223.6	227.0	230.4	233.8	237.1	240.5	243.9	247.3
3	226.1	229.7	233.2	236.7	240.3	243.8	247.4	250.9	254.4	258.0
3 1/16		239.4	243.0	246.7	250.4	254.1	257.8	261.5	265.1	268.8
3⅛		249.2	253.1	256.9	260.7	264.6	268.4	272.2	276.1	279.9
3 3/16			263.3	267.3	271.3	275.3	279.2	283.2	287.2	291.2
3¼			273.7	277.9	282.0	286.2	290.3	294.5	298.6	302.7
3 5/16				288.7	293.0	297.3	301.6	305.9	310.2	314.5
3⅜				299.6	304.1	308.6	313.1	317.5	322.0	326.5
3 7/16					315.5	320.1	324.8	329.4	334.1	338.7
3½					327.1	331.9	336.7	341.5	346.3	351.1
3 9/16						343.8	348.8	353.8	358.8	363.8
3⅝						356.0	361.2	366.3	371.5	376.7
3 11/16							373.8	379.1	384.4	389.8
3¾							386.5	392.0	397.6	403.1

S. A. E. STANDARDS AND RECOMMENDED PRACTICES

(These standards and recommended practices pertaining to engine parts and accessories are printed here only for the information of students as to what was current practice at the time this volume was issued. Those who want to apply S. A. E. standards in new designs should get the latest data directly from the Society of Automotive Engineers in New York City, as standards are revised and new ones added semi-annually.)

CONNECTING-ROD BOLTS



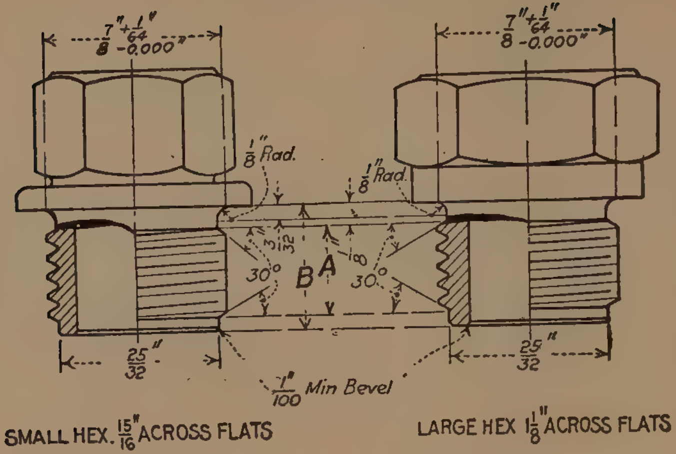
Diam.	A	B	C	D	Threads Per Inch	Pitch Diam.	E¹	F	G No	H Length Minus	R
5/16	Lengths Vary by Even 1/8 Inches, Preferably by 1/4 Inches.	9/16	3/16	0.3125 0.3105	24	0.2854 0.2830	1/2	3/16	48	3/32	0.01 to 1/32
3/8		5/8	7/32	0.3750 0.3730	24	0.3479 0.3455	5/8	7/32	36	9/64	
7/16		11/16	1/4	0.4375 0.4355	20	0.4050 0.4024	11/16	1/4	36	9/64	
1/2		3/4	9/32	0.5000 0.4975	20	0.4675 0.4649	3/4	9/32	36	9/64	1/64 to 3/64
9/16		7/8	5/16	0.5625 0.5600	18	0.5264 0.5234	7/8	5/16	28	3/16	
5/8		1	3/8	0.6250 0.6220	18	0.5889 0.5859	1	3/8	28	3/16	1/32 to 1/16
3/4		1 1/8	7/16	0.7500 0.7470	16	0.7094 0.7062	1 1/8	7/16	28	3/16	
7/8		1 3/8	1/2	0.8750 0.8715	14	0.8286 0.8250	1 1/4	1/2	28	3/16	
1		1 1/2	9/16	1.0000 0.9965	14	0.9536 0.9500	1 1/2	5/8	28	3/16	

Threads are U. S. Form.

[1] Minimum length of usable threads.

Recommended practice for material is S. A. E. Steel No. 2330 or 3130. Heat-treatment should give a Brinell test of from 223 to 285.

SPARK-PLUGS



S. A. E. STANDARD SPARK-PLUG DIMENSIONS

Thread[1]	Hexagon	Thread Length (A)	Skirt Length[2] (B)
7/8–18	15/16	3/8	5/8
7/8–18	15/16	3/8	13/16
7/8–18	15/16	23/32	1
7/8–18	1⅛	3/8	5/8
7/8–18	1⅛	3/8	13/16
7/8–18	1⅛	23/32	1

[1] The outside diameter of the spark-plug threads shall be 0.8750 in. and the number of threads per inch shall be 18. The limits for the pitch diameter shall be 0.8389 in. minimum for the tapped-hole threads. The American form of thread shall be used.

[2] The distance from the gasket seat to the end of the shell.

Firing Points.—The maximum projection of the firing points below the shell shall be 3/16 in.

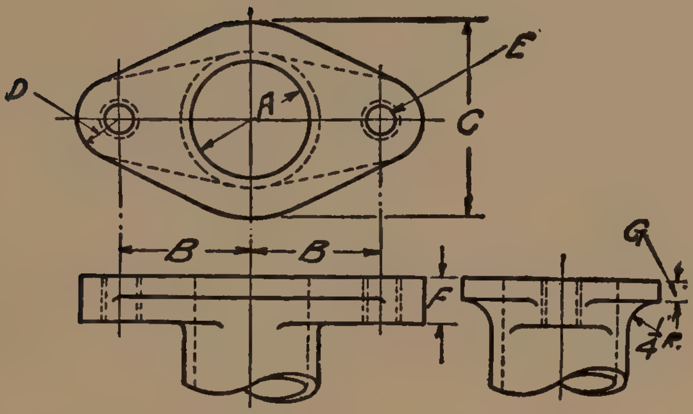
Clearance.—The minimum distance from the spark-plug seat to the nearest object over the spark-plug shall be 2¾ in

ENGINE TRUNNIONS

Type	Diameter, In.	Length, In.
Starting-Crank Bracket	2½ 3	1¼ 1¼
Gear Cover	3¾ 5	1½ 1½

CARBURETER FLANGES

TWO-BOLT TYPE *



Nom. Carbureter Size	A	B	C	D	E[1] Diam.	E[1] Threads Per Inch	F[2]	G[2]
1/2	13/16	29/32	1 7/16	9/32	1/4	20	5/16	3/32
5/8	13/16	29/32	1 7/16	9/32	1/4	20	5/16	3/32
3/4	1 1/16	1 1/8	1 3/4	11/32	5/16	18	3/8	1/8
7/8	1 1/16	1 1/8	1 3/4	11/32	5/16	18	3/8	1/8
1	1 3/16	1 3/16	1 7/8	11/32	5/16	18	13/32	5/32
1 1/4	1 7/16	1 11/32	2 3/16	13/32	3/8	16	15/32	3/16
1 1/2	1 11/16	1 15/32	2 1/2	13/32	3/8	16	15/32	3/16
1 3/4	1 15/16	1 21/32	2 13/16	15/32	7/16	14	9/16	7/32
2	2 3/16	1 25/32	3 1/8	15/32	7/16	14	9/16	7/32

[1] U. S. Standard thread. All dimensions in inches.

[2] *Cast-Iron Carbureter Flanges.*—Flange dimensions *F* and *G* shall be increased ⅛ in. for cast-iron carbureters.

Side-Outlet Carbureter Flanges.—The standard flange dimensions shall be used with the long diameter of the flange in a *vertical* plane when attaching carbureters of the side-outlet type.

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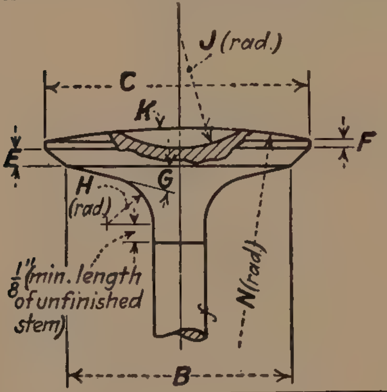
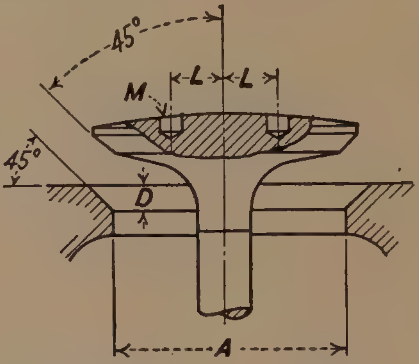
* For specifications of four-bolt-type carburetor flanges see page 683.

POPPET VALVES

Although the accompanying specifications were formulated primarily for steel poppet valves, they are equally applicable to cast-iron valves except that it may be desirable to have slightly more metal under the heads of the latter.





Nom. Size A	B	C	D	E	F	G (deg.)	H	J	K Width	K Depth	L	M Diam.	M Depth	Rad. N
1	31/32	1 5/32	1/16	3/32	1/32	7	1/4	3/4	1/16	3/32	1/4	1/8	1/8	5
1 1/8	1 3/32	1 9/32	1/16	3/32	1/32	7	1/4	3/4	1/16	3/32	1/4	1/8	1/8	5
1 1/4	1 7/32	1 13/32	1/16	3/32	1/32	7	1/4	1	1/16	3/32	1/4	1/8	1/8	5
1 3/8	1 11/32	1 17/32	1/16	3/32	1/32	7	1/4	1	1/16	3/32	3/8	1/8	1/8	5
1 1/2	1 15/32	1 3/4	3/32	9/64	3/64	7	3/8	1 1/4	1/16	3/32	3/8	1/8	1/8	6
1 5/8	1 19/32	1 7/8	3/32	9/64	3/64	7	3/8	1 1/4	1/16	3/32	3/8	1/8	1/8	6
1 3/4	1 23/32	2	3/32	9/64	3/64	7	3/8	1 1/4	3/32	3/32	3/8	1/8	1/8	6
1 7/8	1 27/32	2 1/8	3/32	9/64	3/64	7	3/8	1 1/4	3/32	3/32	3/8	1/8	1/8	6
2	1 31/32	2 5/16	1/8	11/64	1/16	12	7/16	1 1/4	3/32	3/32	3/8	1/8	1/8	7
2 1/8	2 3/32	2 7/16	1/8	11/64	1/16	12	7/16	1 1/2	3/32	7/64	1/2	5/32	5/32	7
2 1/4	2 7/32	2 9/16	1/8	11/64	1/16	12	7/16	1 1/2	3/32	7/64	1/2	5/32	5/32	7
2 3/8	2 11/32	2 11/16	1/8	11/64	1/16	12	7/16	1 1/2	3/32	7/64	1/2	5/32	5/32	8
2 1/2	2 15/32	2 29/32	5/32	7/32	1/16	12	9/16	1 1/2	3/32	1/8	1/2	5/32	5/32	8
2 5/8	2 19/32	3 1/32	5/32	7/32	1/16	12	9/16	1 1/2	3/32	1/8	5/8	3/16	3/16	8
2 3/4	2 23/32	3 5/32	5/32	7/32	5/64	12	9/16	1 1/2	3/32	5/32	5/8	3/16	3/16	9
2 7/8	2 27/32	3 9/32	5/32	7/32	5/64	12	9/16	1 1/2	3/32	5/32	5/8	3/16	3/16	9
3	2 31/32	3 13/32	5/32	7/32	5/64	12	9/16	1 1/2	3/32	5/32	5/8	3/16	3/16	9

All dimensions in inches.

Concentricity.—The concentricity of the valve-seats with the ground stems shall be inspected by rolling the valve in a V-block, the length of which shall be the length of the ground stem, and which shall be relieved at the center, this relief extending to not more than ½ in. from each end. The valves are to be held lengthwise by a ball bearing against the center of the tip of the stem. The indicator pin shall bear against the valve-seat at an angle of 90 deg.

When so inspected the total variation in indicator reading shall not exceed 0.002 in., that is, the valve seats must be concentric with stems within 0.001 in.

Taper of Stems.—The maximum out-of-roundness or taper of valve stems shall not exceed the tolerance of the valve stem diameters.

Valve-Stem Tips.—Valve-stem tips shall be hardened to give a scleroscope reading of from 55 to 65 after finishing. It is recommended that the length of tips from the end of the stems to the stem-grooves be not less than the finished diameter of the stems so as to prevent undue breakage after hardening.

Spring Retainers.—The type of spring-retainer mechanism used shall be optional, but the operations performed on valve-stems shall have the same machining limits as specified for the tips in the above specifications.

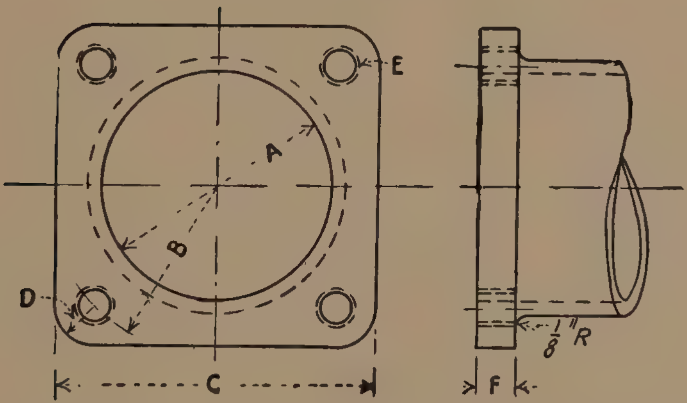
Tolerances.—The tolerances on fractional dimensions shall be plus and minus 0.010 in.

Valve-Head Slots.—In order to facilitate the regrinding of valves and valve-seats, either a slot shall be milled, or holes shall be drilled in the valve-head as shown in the accompanying illustrations.

(Continued on page 684)

CARBURETOR FLANGES

FOUR-BOLT TYPE

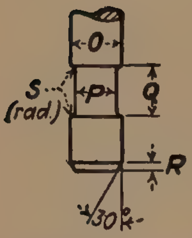


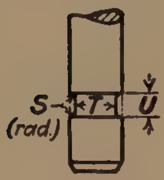
Nom. Carb. Size	A	B	C	D	E[1]		F
					Diam.	Threads Per Inch	
$2\frac{1}{2}$	$2\frac{11}{16}$	$2\frac{1}{32}$	$3\frac{3}{4}$	$\frac{7}{16}$	$\frac{3}{8}$	16	$\frac{7}{16}$
3	$3\frac{3}{16}$	$2\frac{5}{16}$	$4\frac{1}{4}$	$\frac{1}{2}$	$\frac{7}{16}$	14	$\frac{1}{2}$
$3\frac{1}{2}$	$3\frac{11}{16}$	$2\frac{5}{8}$	$4\frac{7}{8}$	$\frac{9}{16}$	$\frac{1}{2}$	13	$\frac{9}{16}$

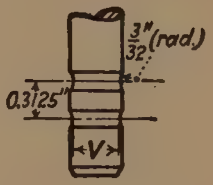
[1] U. S. Standard thread. All dimensions in inches.

POPPET VALVES

(*Continued*)







A	O		P		Q		T		U		V	
	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
1	0.3100	0.3090	0.250	0.245	0.328	0.322	0.235	0.230	0.144	0.138	0.250	0.245
1⅛	0.3100	0.3090	0.250	0.245	0.328	0.322	0.235	0.230	0.144	0.138	0.250	0.245
1¼	0.3100	0.3090	0.250	0.245	0.328	0.322	0.235	0.230	0.144	0.138	0.250	0.245
1⅜	0.3100	0.3090	0.250	0.245	0.328	0.322	0.235	0.230	0.144	0.138	0.250	0.245
1½	0.3725	0.3715	0.309	0.303	0.391	0.385	0.253	0.247	0.144	0.138	0.309	0.303
1⅝	0.3725	0.3715	0.309	0.303	0.391	0.385	0.253	0.247	0.144	0.138	0.309	0.303
1¾	0.3725	0.3715	0.309	0.303	0.391	0.385	0.253	0.247	0.144	0.138	0.309	0.303
1⅞	0.3725	0.3715	0.309	0.303	0.391	0.385	0.253	0.247	0.144	0.138	0.309	0.333
2	0.4345	0.4335	0.312	0.306	0.516	0.510	0.312	0.306	0.175	0.169	0.372	0.366
2⅛	0.4345	0.4335	0.312	0.306	0.516	0.510	0.312	0.306	0.175	0.169	0.372	0.366
2¼	0.4345	0.4335	0.312	0.306	0.516	0.510	0.312	0.306	0.175	0.169	0.372	0.366
2⅜	0.4345	0.4335	0.312	0.306	0.516	0.510	0.312	0.306	0.175	0.169	0.372	0.366
2½	0.5595	0.5575	0.437	0.430	0.578	0.572	0.405	0.398	0.206	0.200	0.495	0.489
2⅝	0.5595	0.5575	0.437	0.430	0.578	0.572	0.405	0.398	0.206	0.200	0.495	0.489
2¾	0.5595	0.5575	0.437	0.430	0.578	0.572	0.405	0.398	0.206	0.200	0.495	0.489
2⅞	0.5595	0.5575	0.437	0.430	0.578	0.572	0.405	0.398	0.206	0.200	0.495	0.489
3	0.5595	0.5575	0.437	0.430	0.578	0.572	0.405	0.398	0.206	0.200	0.495	0.489

Dimensions for *R* are $\frac{1}{32}$ in. for diameters from 1 to 2⅜ in. and $\frac{1}{16}$ in. for diameters from 2½ to 3 in.

Dimensions for *S* are 0.008 to 0.012 in. for diameters from 1 to 1⅞ in., 0.010 to 0.015 in. for diameters from 2 to 2⅜ in., and 0.015 to 0.020 in. for diameters from 2½ to 3 in.

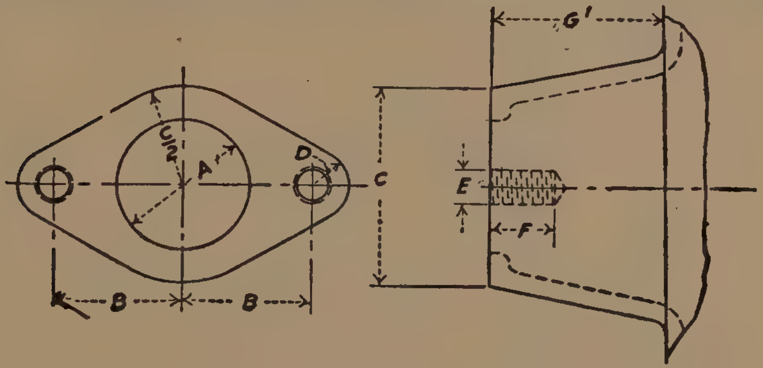
Valve Lengths.—The over-all lengths of valves shall be measured from the large diameter of the valve-seats to the tip of the stems and shall vary in increments of ⅛ in. on valves up to and including the 2-in. size and ¼ in. from 2⅛ to 3-in. sizes.

Grinding of Stems.—The ground length of stems shall be shown on valve drawing. The specified stem-diameter limits shall be measured on and applied to only the ground portion of stems.

RADIATORS

CAST RADIATORS

Fittings.—Radiator inlet and outlet fittings shall be cast separate from the radiator tanks. The finished face of the flanges cast on the radiator tanks shall extend ⅛ in. beyond any other projection on the tanks.



Nominal Flange Diameter (A)	B	C	D	E		F
				Diam.	Thrds. per in.	
1 3/16	1 3/16	1 7/8	11/32	5/16	18	5/8
1 7/16	1 11/32	2 3/16	13/32	3/8	16	3/4
1 11/16	1 15/32	2 1/2	13/32	3/8	16	3/4
1 15/16	1 21/32	2 13/16	15/32	7/16	14	7/8
2 3/16	1 25/32	3 1/8	15/32	7/16	14	7/8

[1] Dimension G: 1⅛ in. for top tanks for motor trucks of capacities under 3½ tons, 1⅝ in. for top tanks for motor trucks of capacities of 3½ tons or over, and 1 in. for bottom tanks for all motor truck capacities.

Capscrew holes shall be blind.

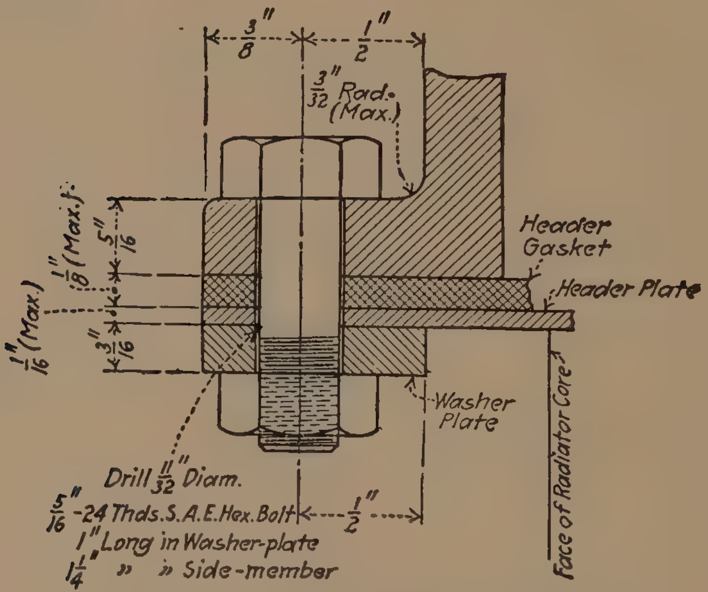
The diameter of the capscrew holes in the water-pipe flanges shall be 1/32 in. larger than the capscrew diameters.

The capscrew shall extend into the flange holes a distance equal to 1½ times their diameter.

RADIATORS

(*Continued*)

Overflow Tubes.—In tubular radiators having internal overflow tubes, the tubes shall be of seamless brass or copper and shall be attached to the bottom tank by ½-in. American Standard taper pipe-thread brass plugs. The jacket tube shall be ⅝ in. diameter and the overflow tube shall be ⅜ in. diameter.



CAST RADIATOR HEADER FLANGE DIMENSIONS

Drains.—The size of the tapped hole for radiator drains shall be ⅜ in. American Standard taper pipe-thread.

Header Flanges.—Bolt-hole centers shall be spaced 2½ in. apart wherever possible.

Cooling Flanges.—Cast radiator tanks shall be designed without ribs or cooling flanges.

Tie-Rod Fittings.—Tie-rod fittings shall be in accordance with the illustration on page A28.

Radiator Tests—A pressure of 5 lb. per sq. in. shall be used in making all tests for radiator leaks.

Header Plates.—Header-plate widths shall be 5, 5½, 6, 6½, and 7 in.

CLUTCH HOUSINGS

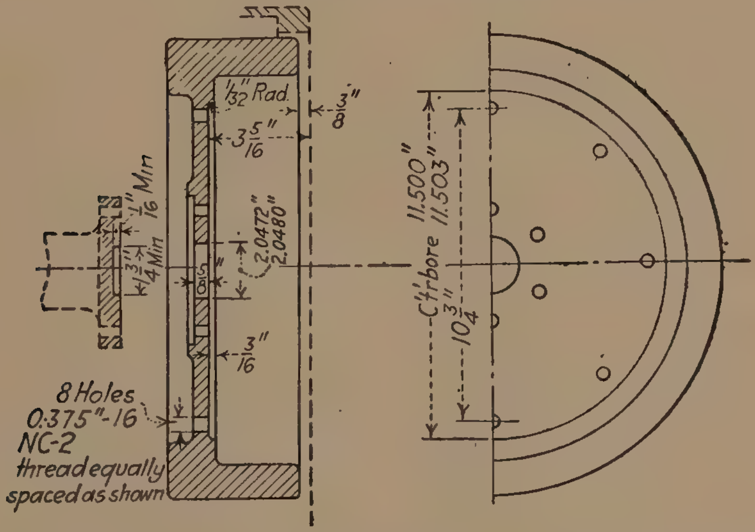
The cap-screw holes in the clutch housing flanges shall be 1/32 in. larger than the nominal diameter of the flywheel housing cap-screws.

The diameter of the pilot on the flange of the clutch housing shall be the same as the nominal diameter of the bore in the flywheel housing; the tolerances shall be plus 0.0000 and minus 0.0050 in.; and the maximum eccentricity shall be 0.0025 in. (Indicator reading, 0.0050 in.)

The maximum variation of the face of the clutch housing flange from its true position, when rotated about its axis, shall be 0.0025 in. (Indicator reading, 0.0050 in.)

The limits for the outside diameter of the clutch driving-member shall be 11.497 and 11.499 in.

DISC-CLUTCH TYPE FLYWHEELS



Fixed dimensions do not apply to flywheels for the No. 5 housing.

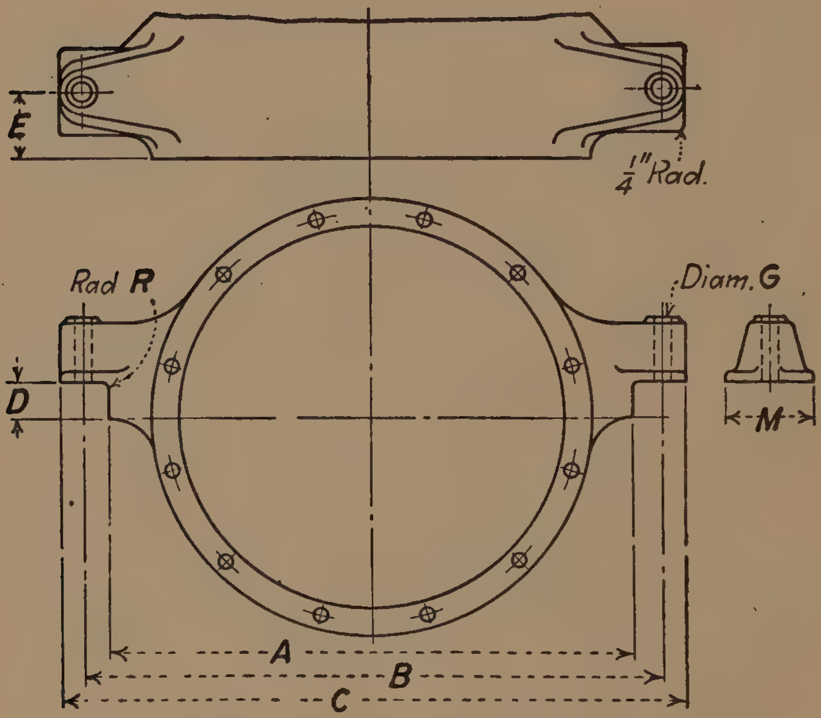
The maximum variation of the counterbored surface of the disc-clutch type of flywheel from its true position, when the flywheel is rotated on its axis, shall be 0.0025 in. when measured at a radius of 5½ in. (Indicator reading, 0.0050 in.)

The maximum eccentricity for the 11½-in. counterbore in the disc-clutch type of flywheel shall be 0.0025 in. (Indicator reading, 0.0050 in.)

The eccentricity for the bore in the disc-clutch type of flywheel for the pilot bearing shall be 0.0025 in. (Indicator reading, 0.0050 in.)

The maximum distance from the flywheel face to the ends of the crankshaft flywheel bolts shall be 11/16 in.

ENGINE REAR SUPPORT ARMS



Size No.	A	B	C	D	E	G	R Max.	M
1	23½	26¾	28¾	2	3	25/32	7/16	4
2	23½	26¾	28¾	2½	3	25/32	⅜	3½
3	23⅛	24½	25¾	2½	3	21/32	¼	3½
4	23⅛	24½	25¾	2½	3	21/32	¼	2¾
5	23⅛	24½	25¾	2½	3	21/32	¼	2¾
2T[1]	20	21¾	23½	3	3	21/32	¼	4

[1] Size No. 2T is intended primarily for tractor engines.

TRACTOR ENGINE FLYWHEEL HOUSINGS.

Flywheel housings No. 1 and 2, page A1, shall be used for tractor practice.

When it is desired to use ½ in.-13 in place of ⅜ in.-16 S. A. E. Coarse cap-screws, they shall be located on the standard bolt-circle. The outside diameter of the clutch housing flange may be increased ½ in., but the inside diameter shall remain the same.

If the clutch housing flange is unfinished, the outside diameter shall be increased ½ in., but the inside diameter and the bolt-circle diameter shall remain the same.

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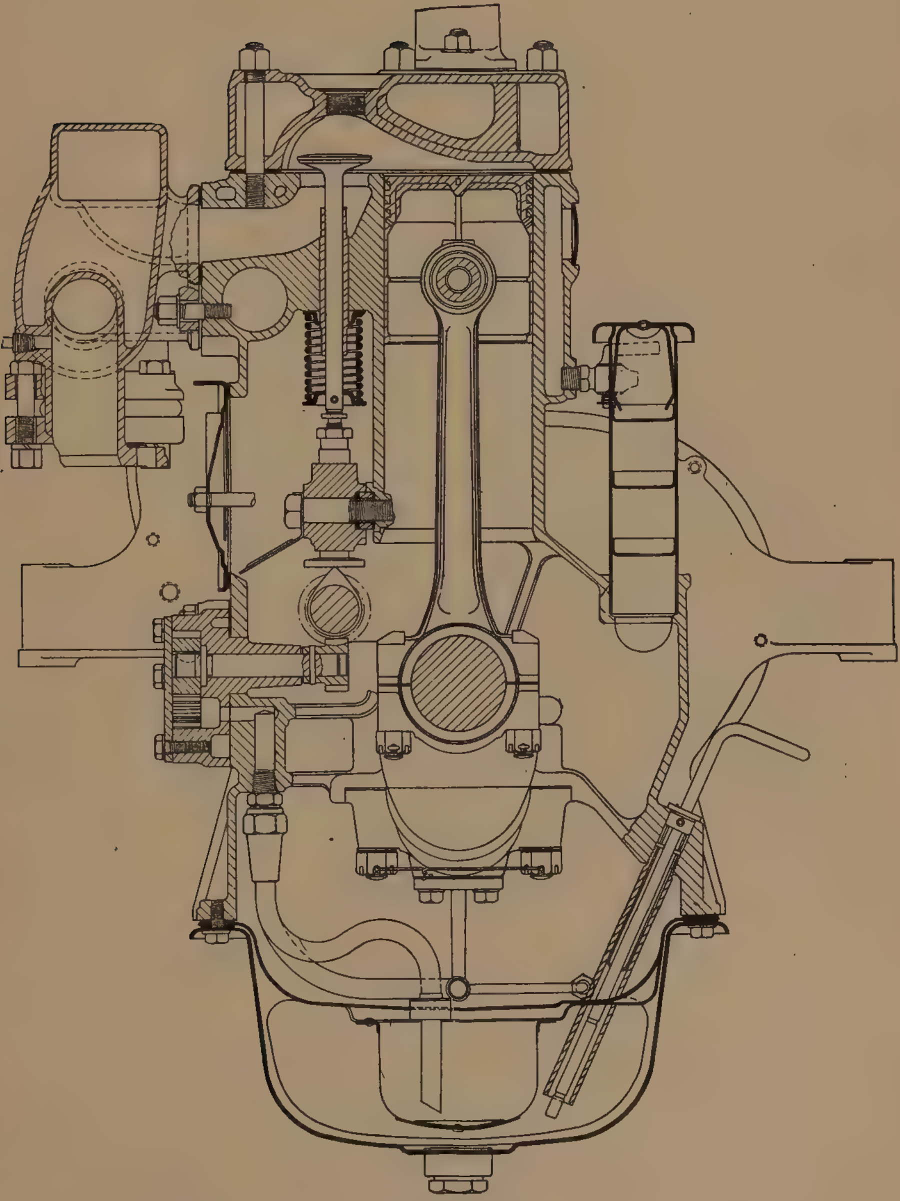
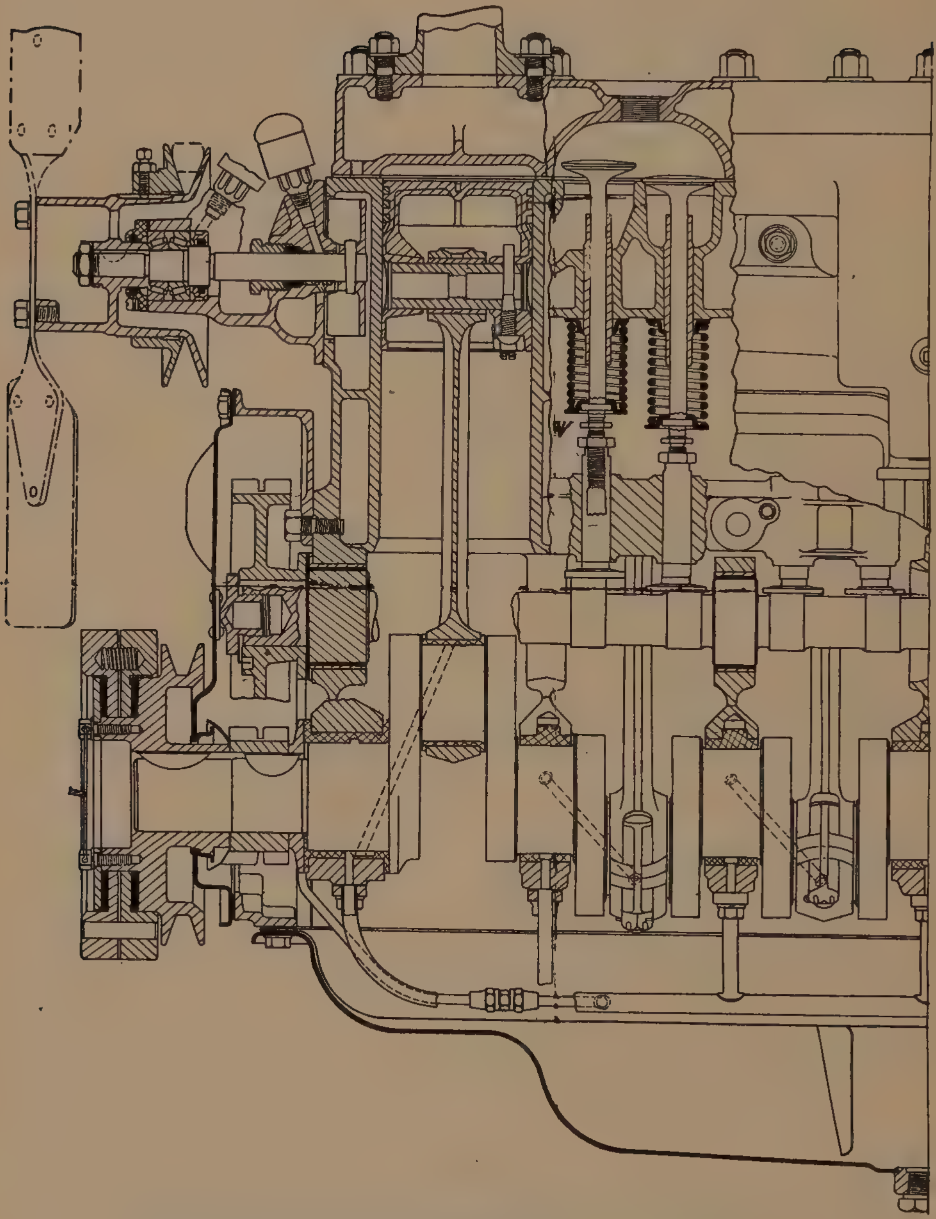
PLATE SUPPLEMENT



Peerless Six-Cylinder Engine, 3¼ by 4⅝ In.



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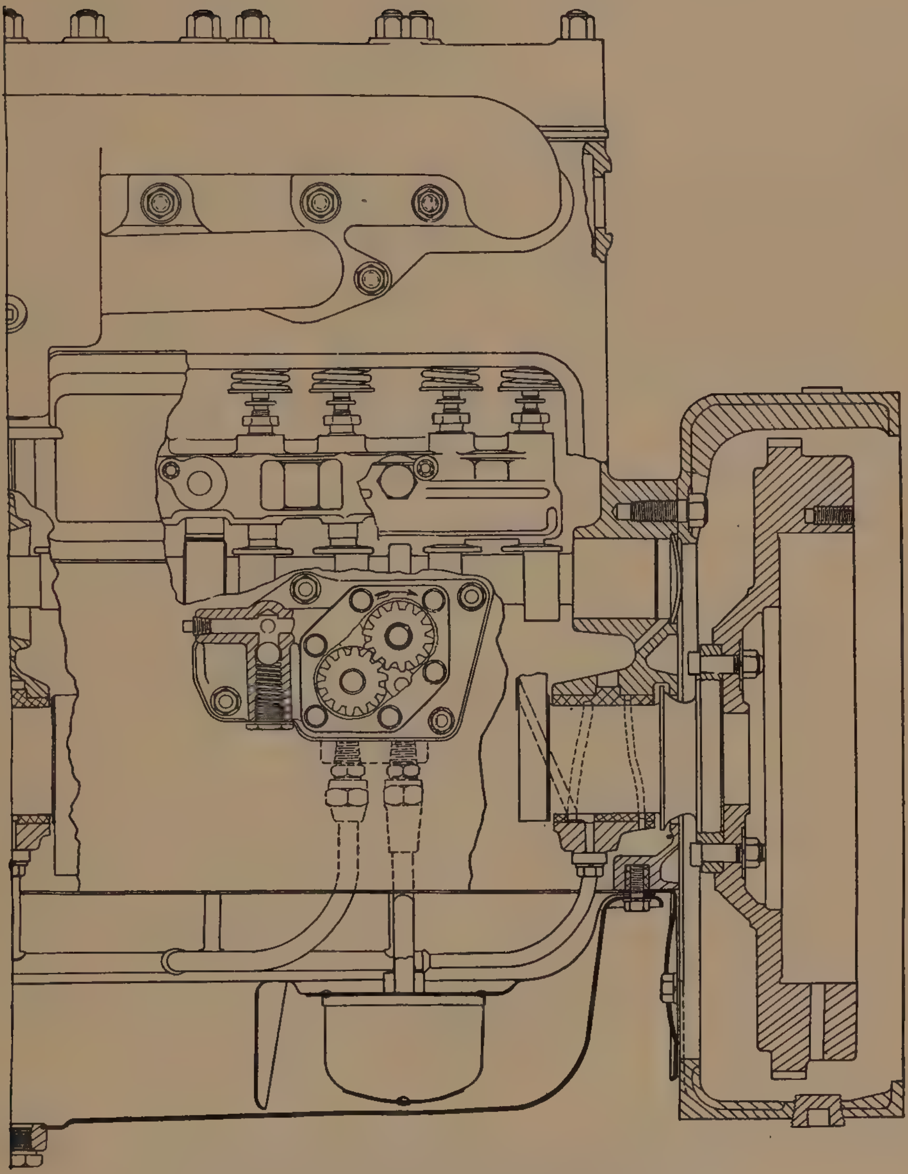

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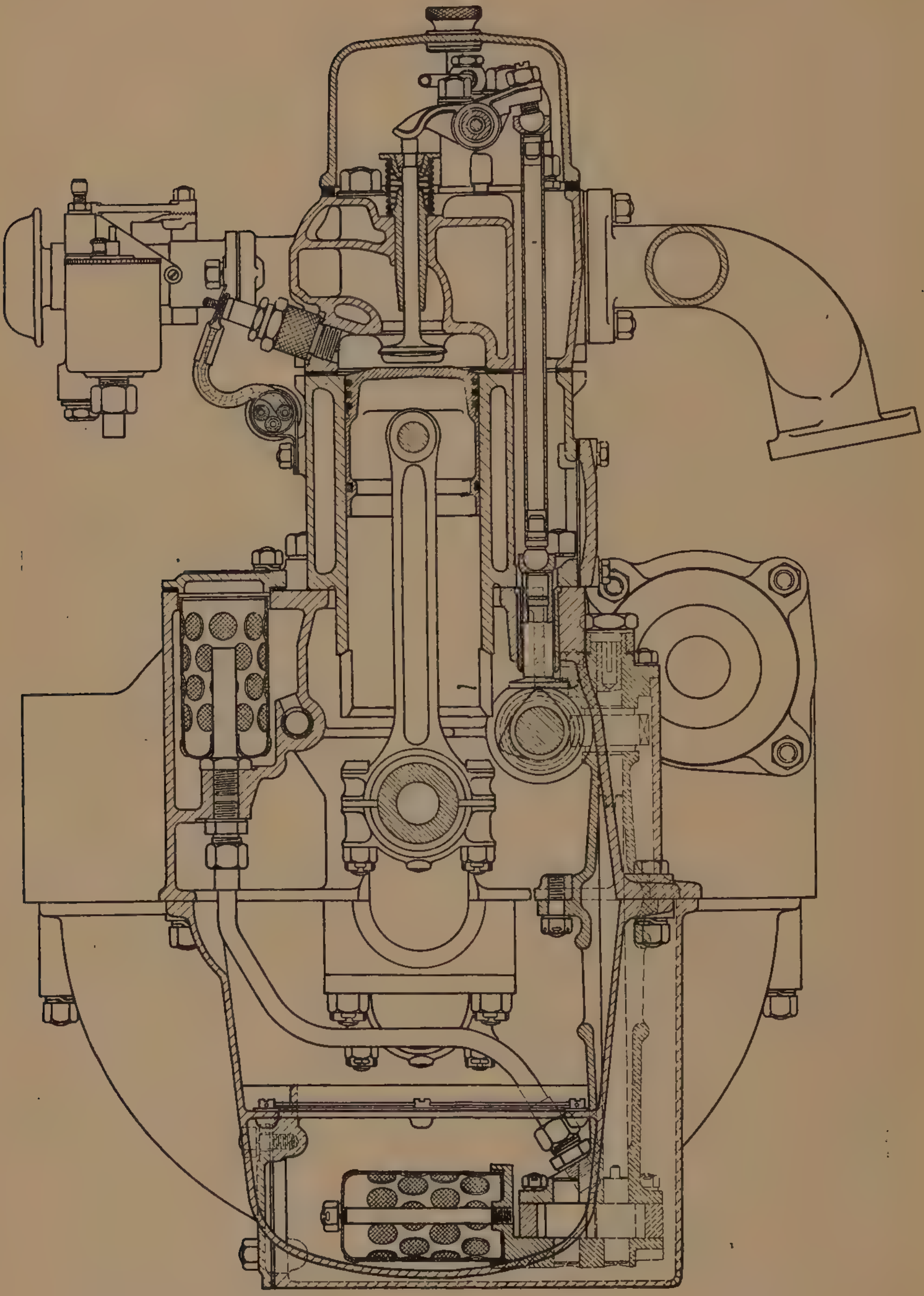
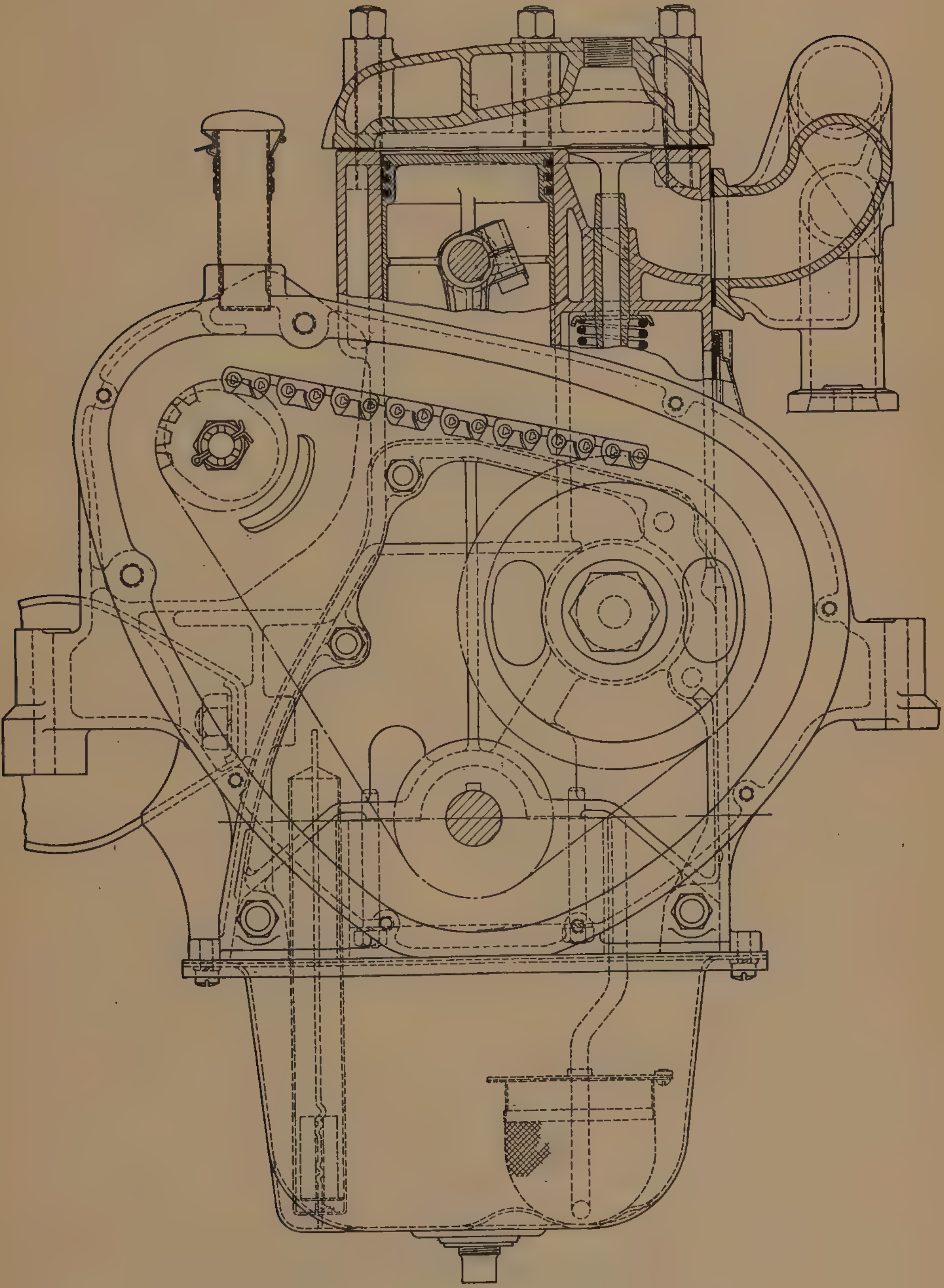
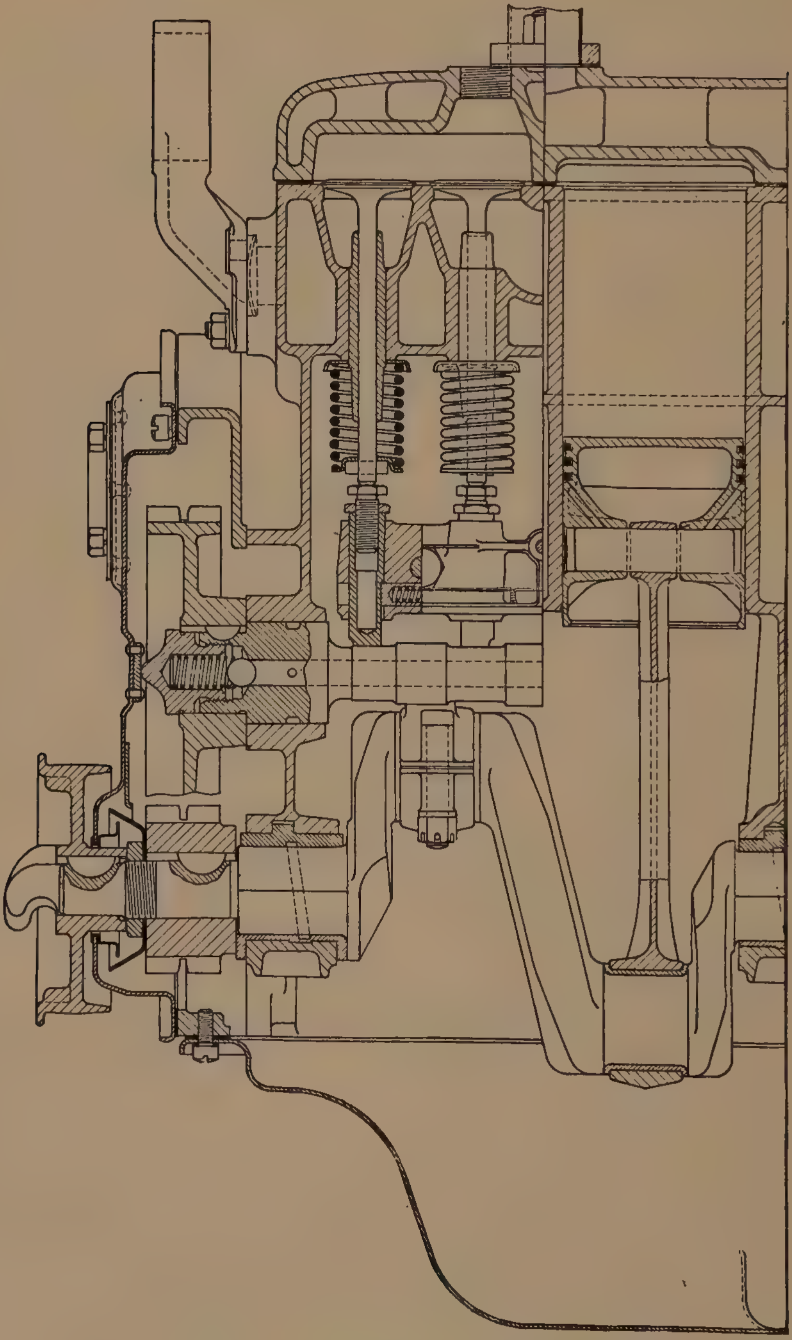


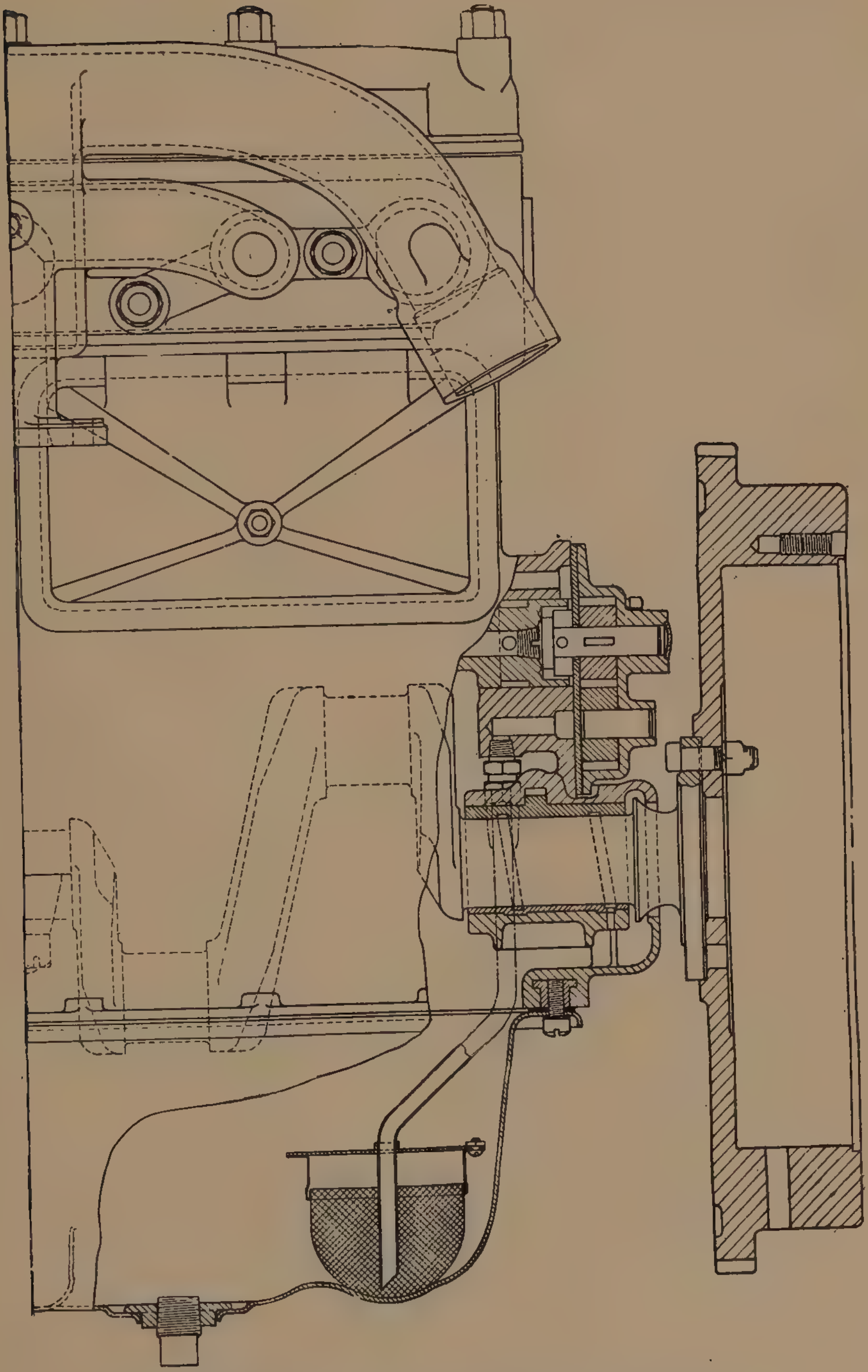
MEADOWS (BRITISH) FOUR-CYLINDER ENGINE, 2.68 BY 3.94 IN.



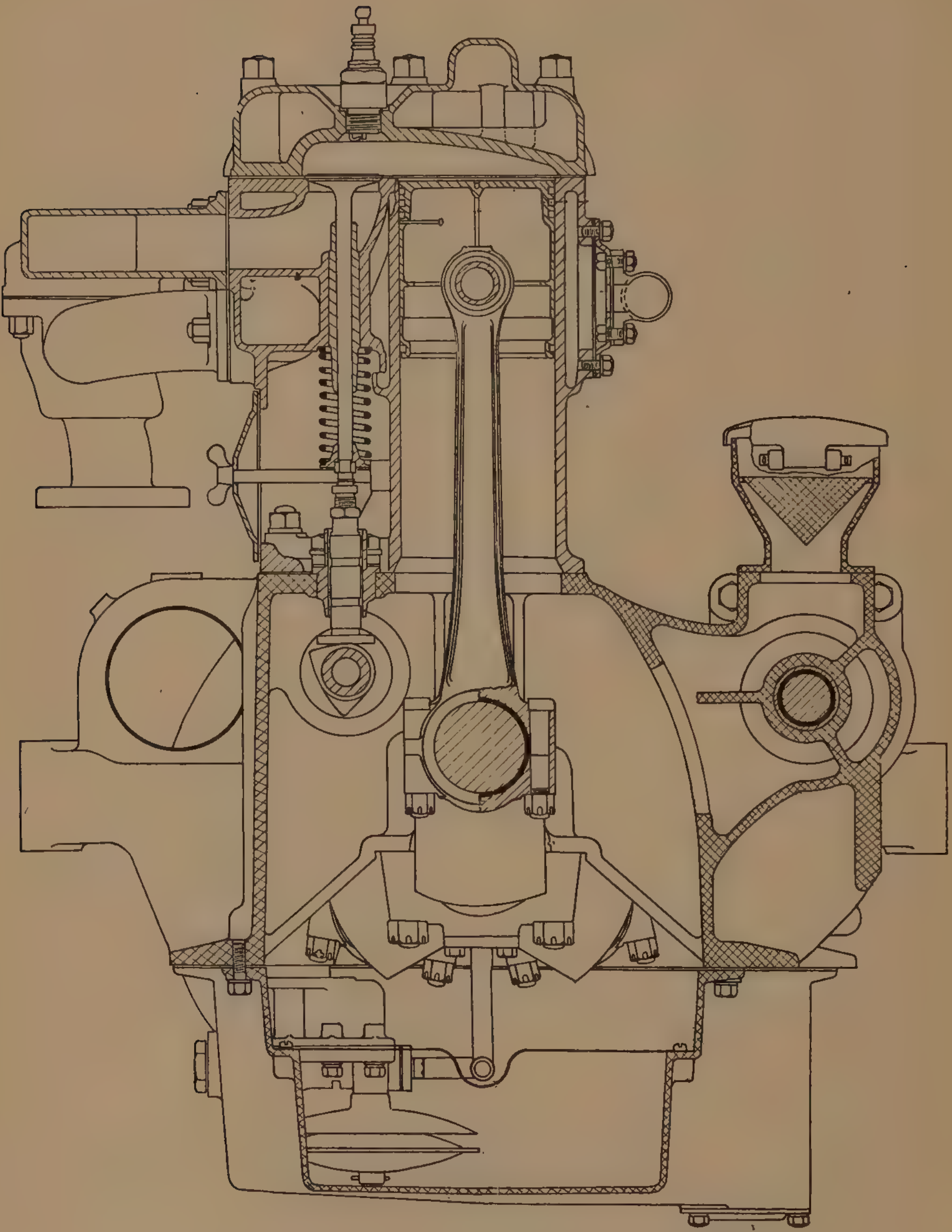
STAR FOUR-CYLINDER ENGINE, 3⅜ BY 4¼ IN.



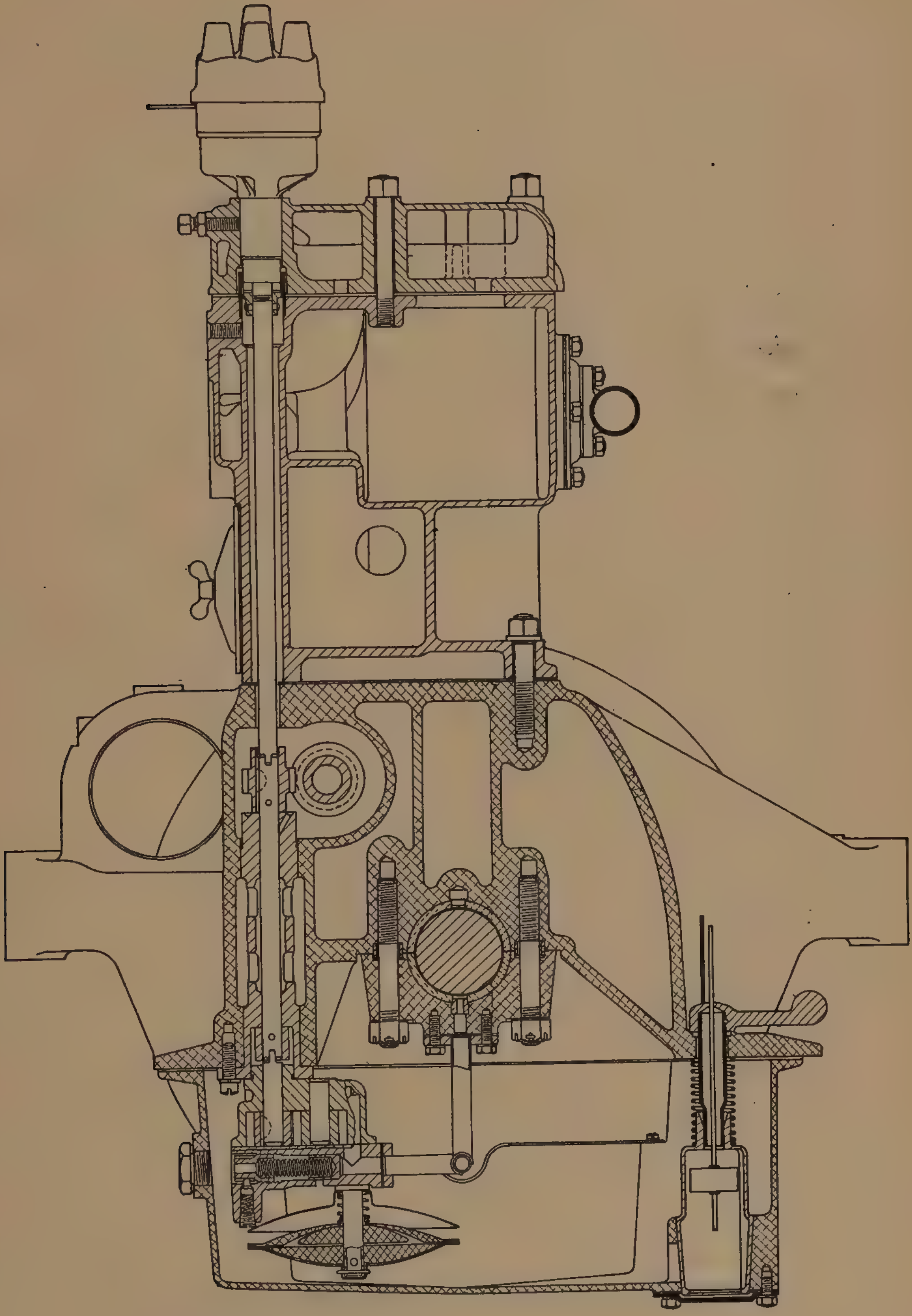
STAR FOUR-CYLINDER ENGINE, 3⅜ BY 4¼ IN.



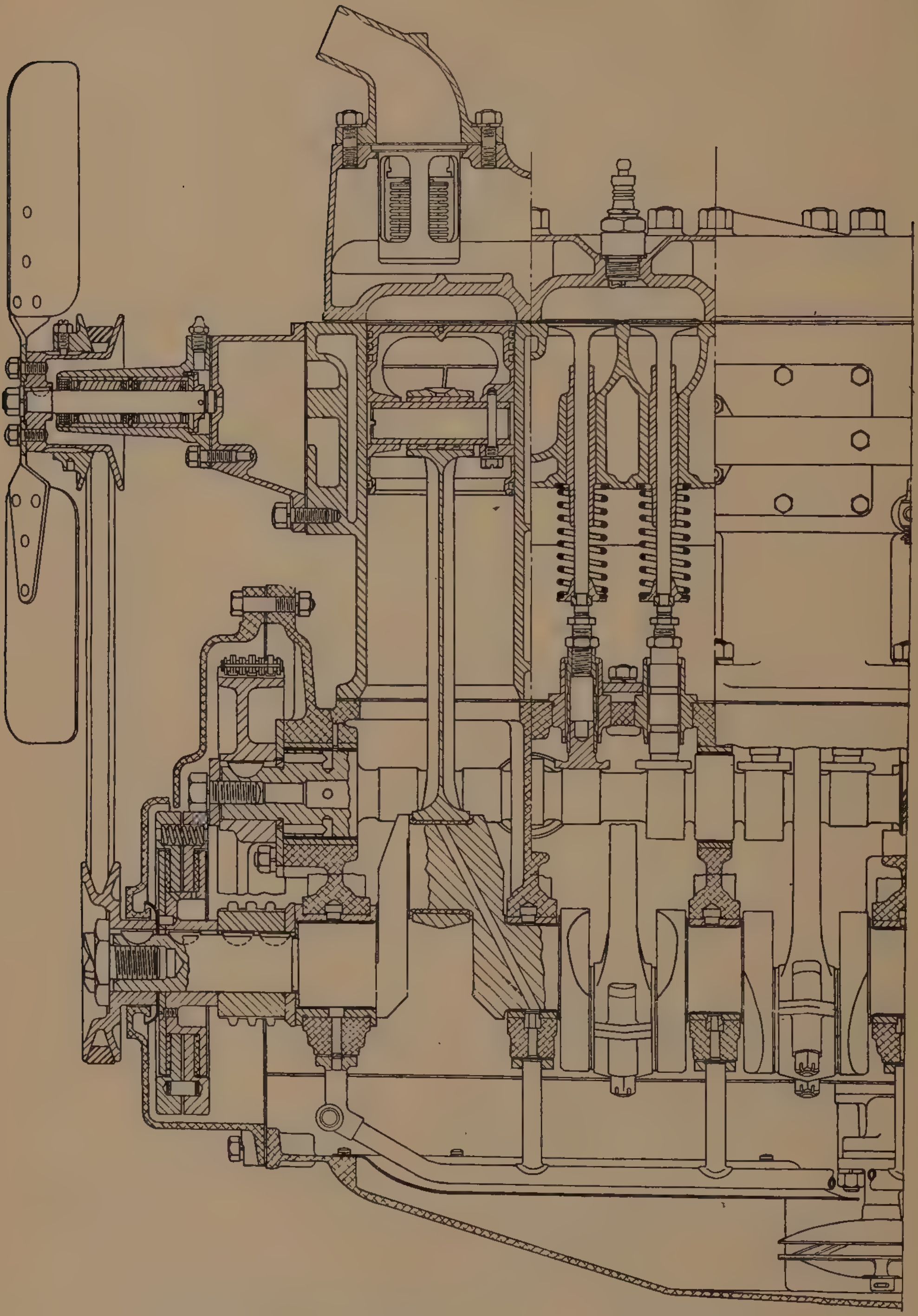
Star Four-Cylinder Engine, 3⅜ by 4¼ In.



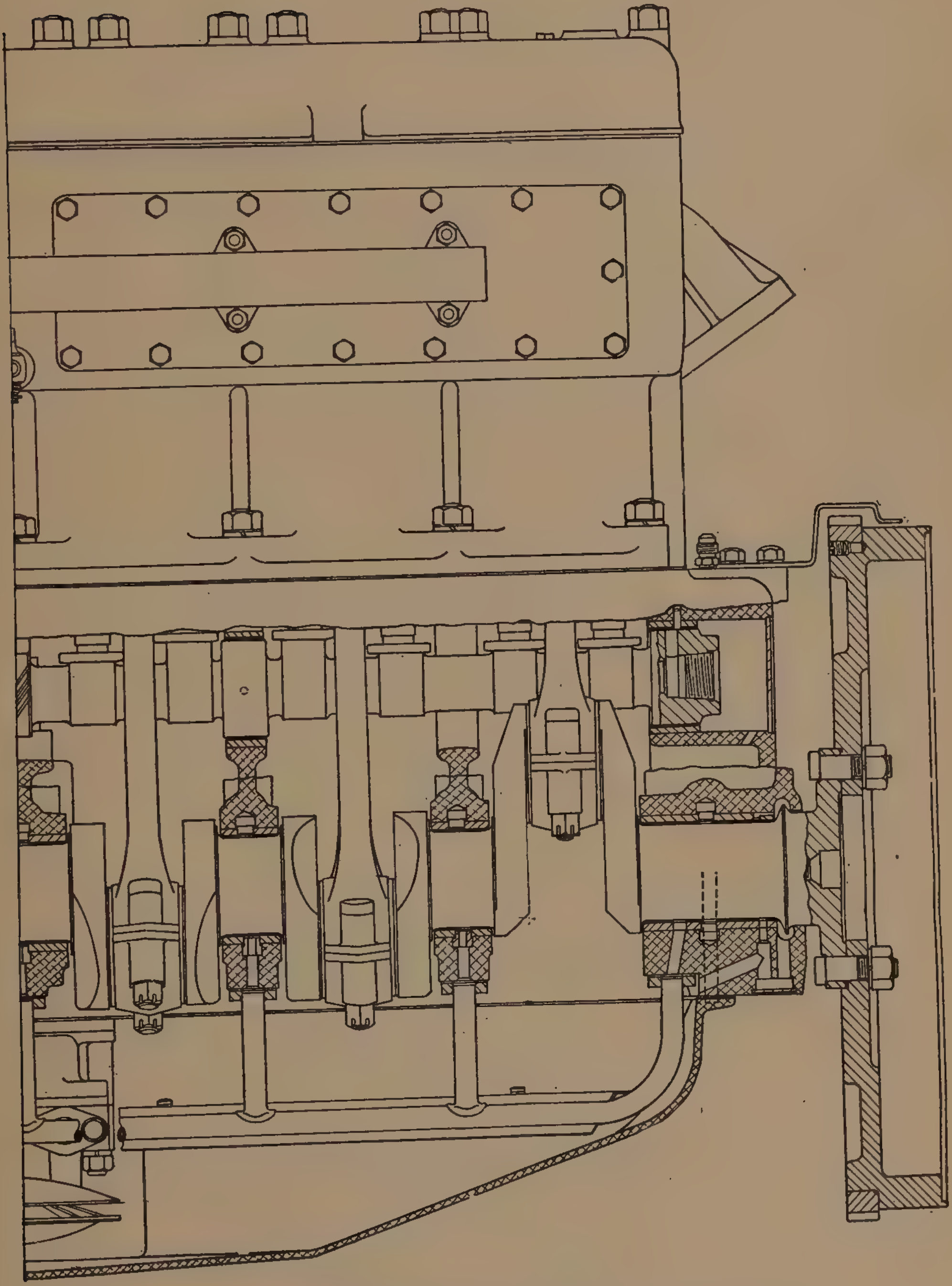
LOCOMOBILE SIX-CYLINDER ENGINE, 3⅞ BY 5¼ IN.



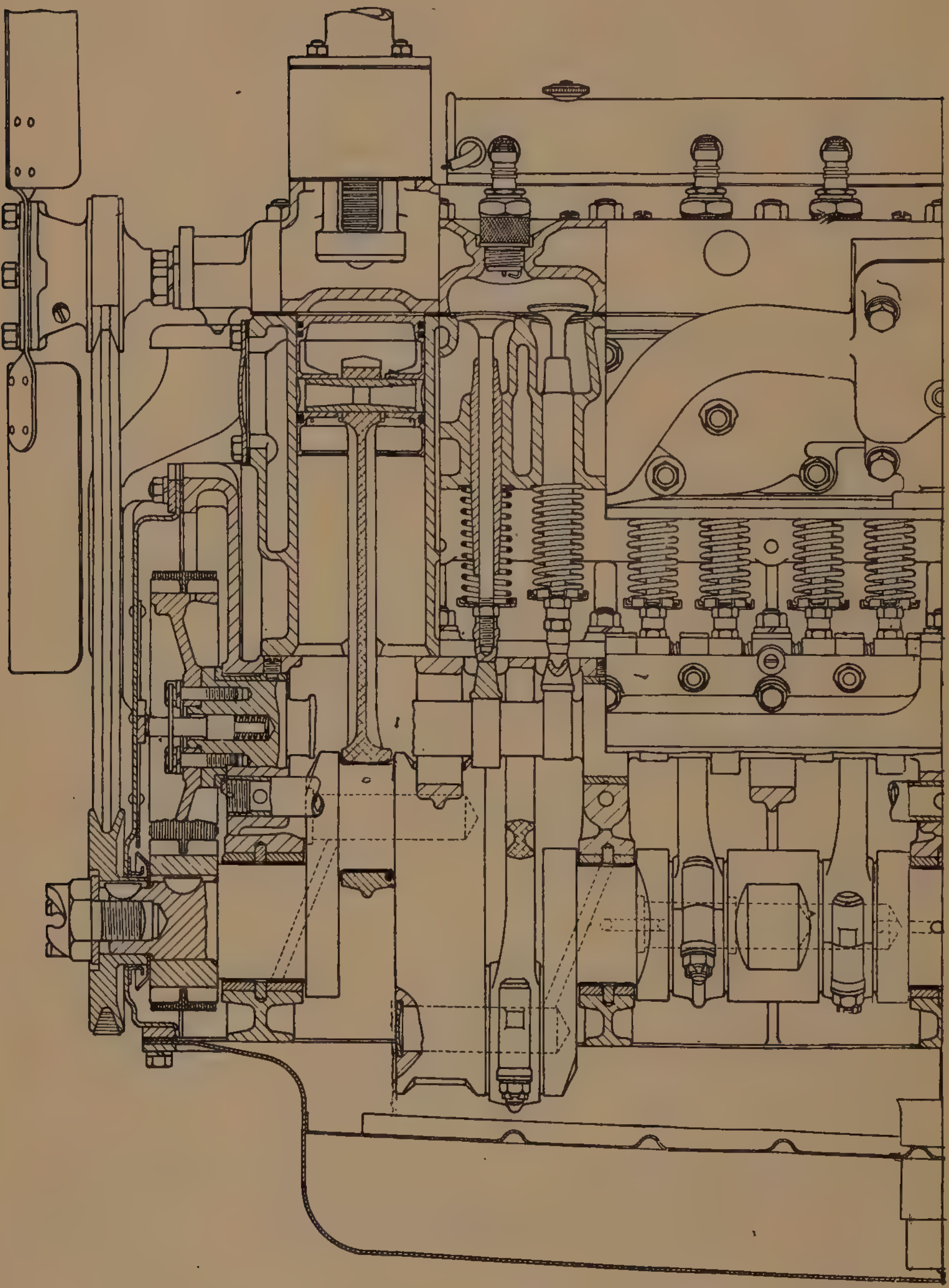
LOCOMOBILE SIX-CYLINDER ENGINE, 3⅞ BY 5¼ IN.



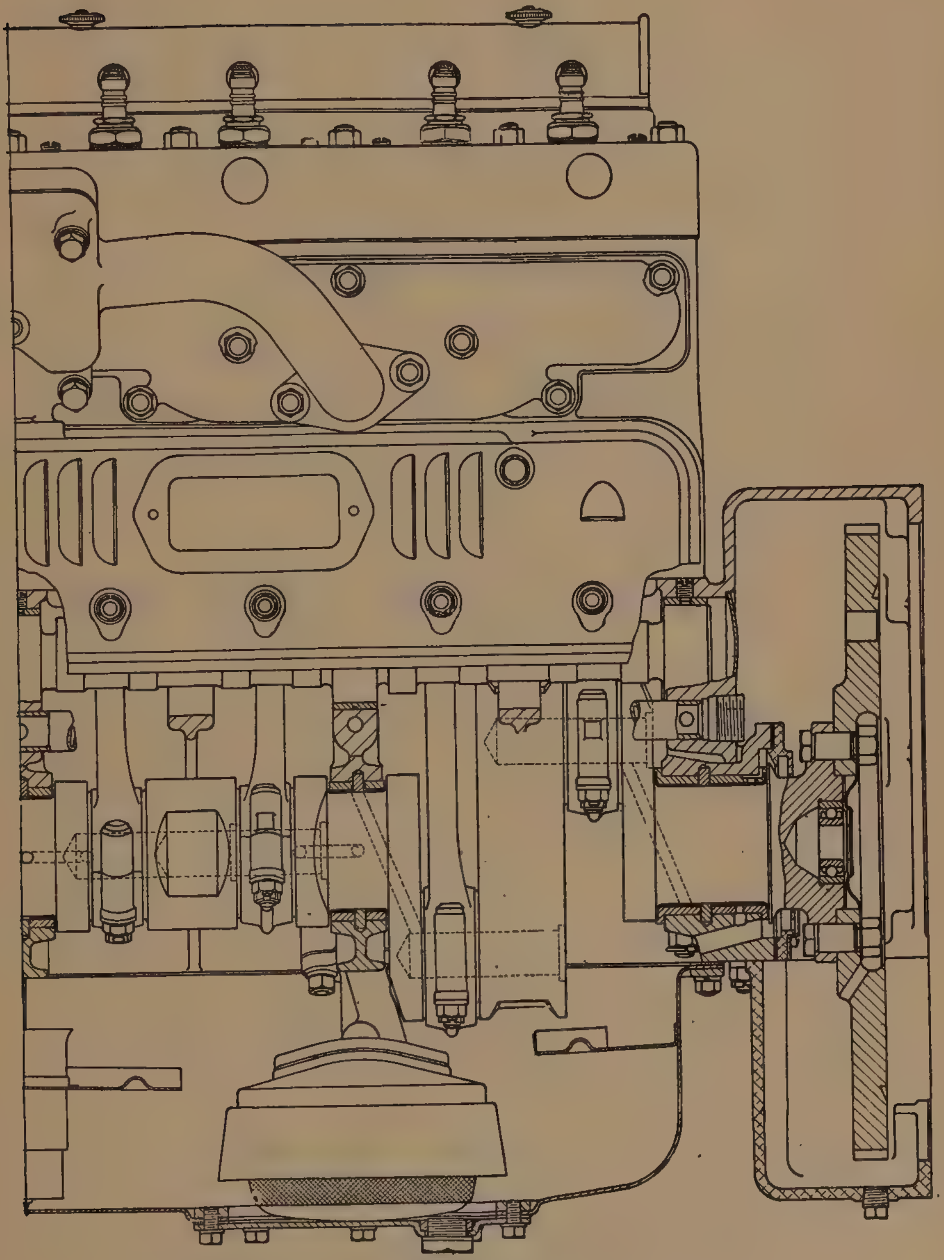
Locomobile Six-Cylinder Engine, 3⅞ by 5¼ In.



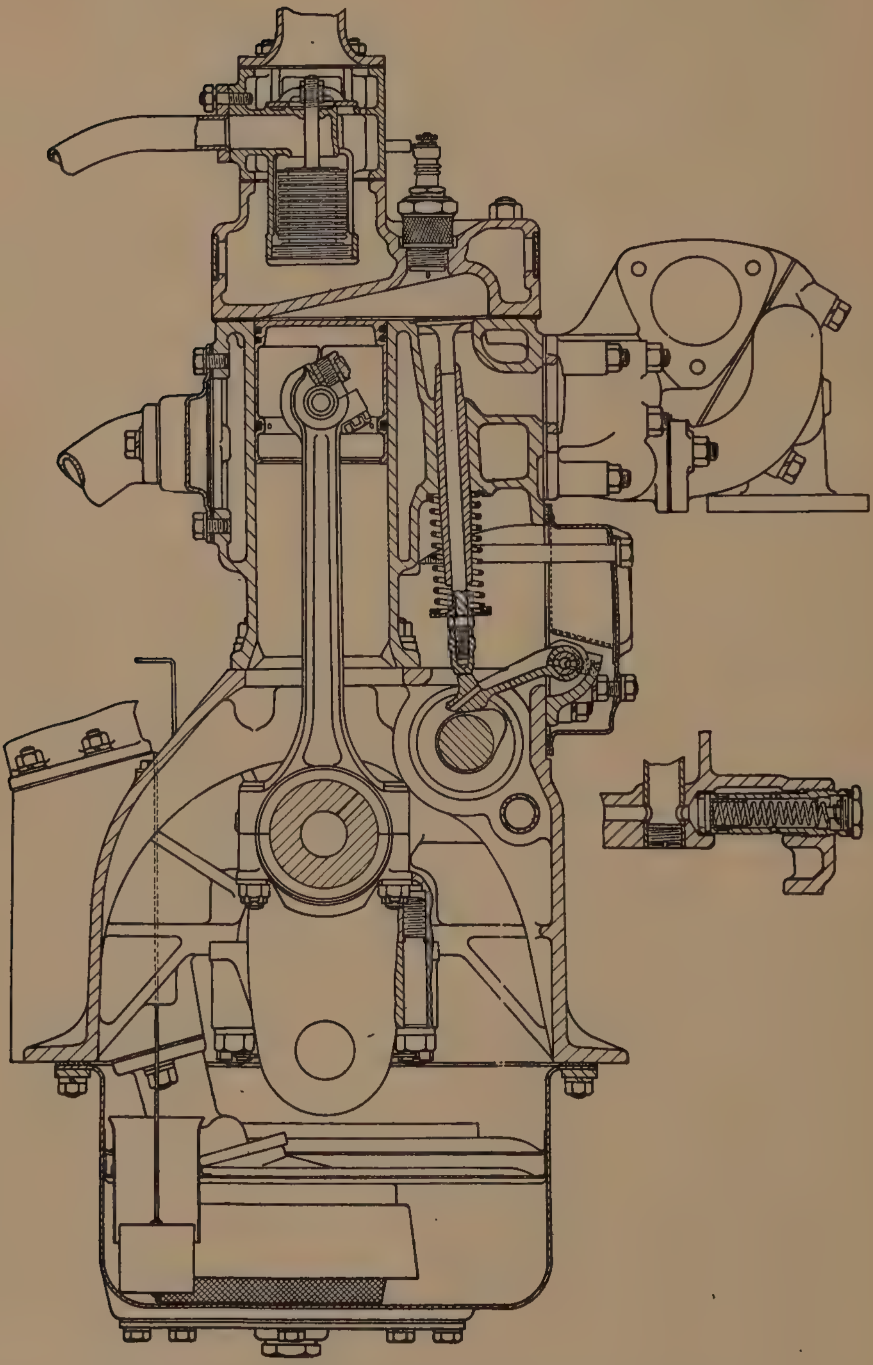
Locomobile Six-Cylinder Engine, 3⅞ by 5¼ In.



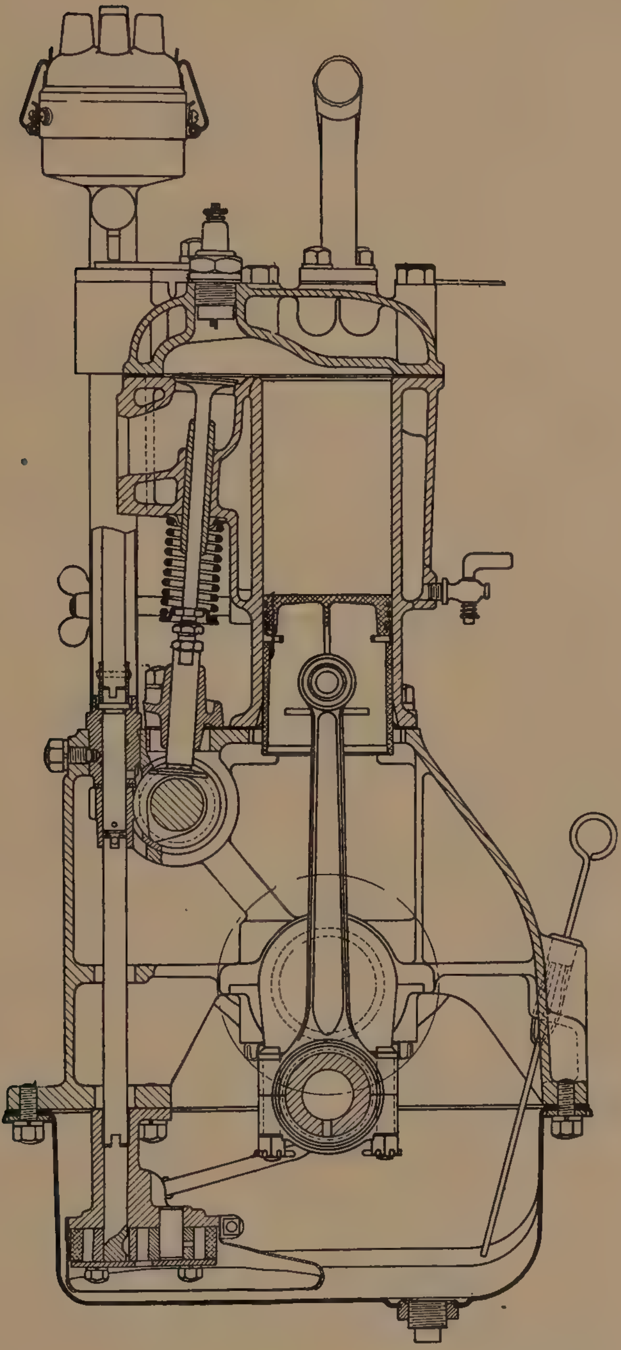
Hupmobile Eight-Cylinder Engine, 3 by 4¾ In.



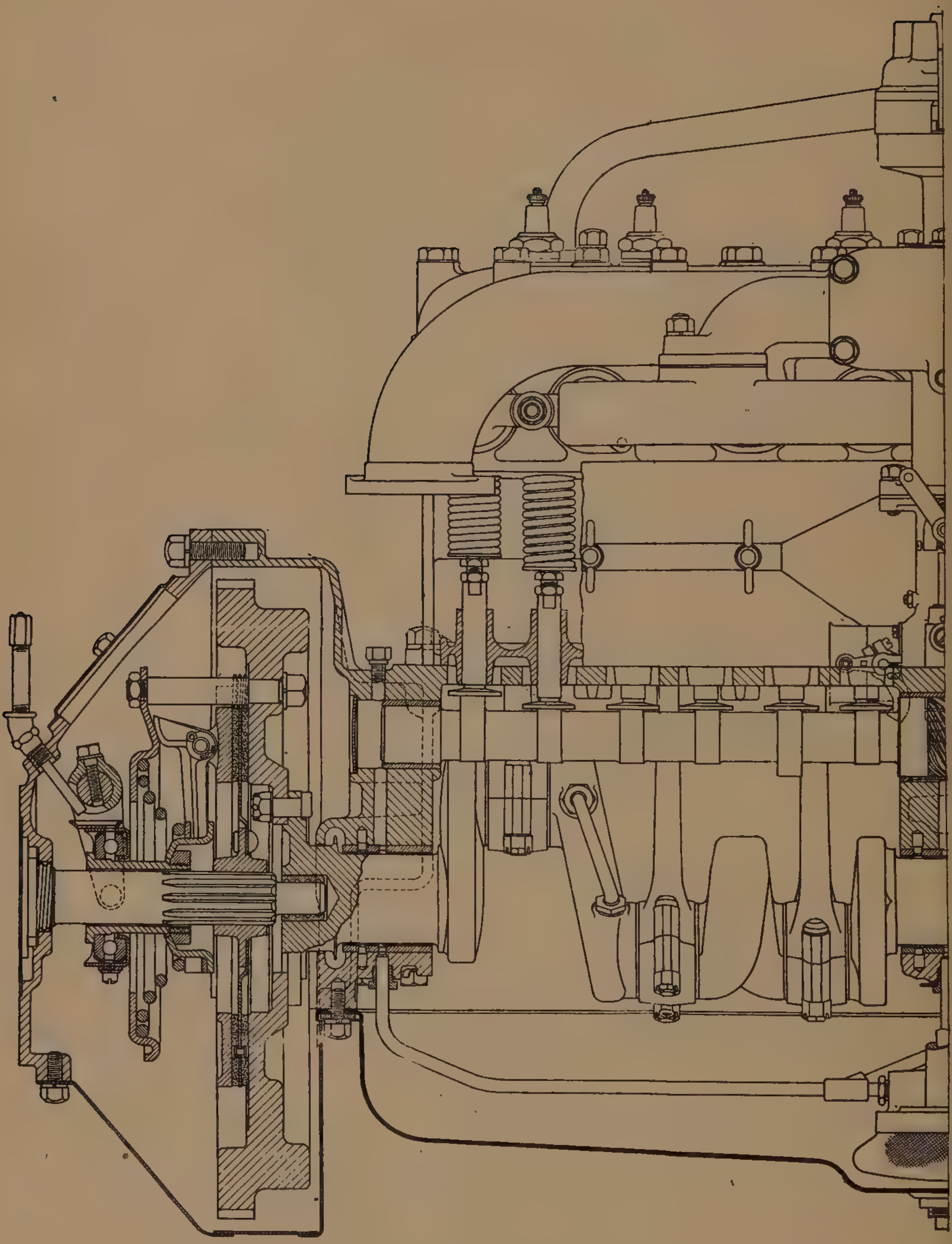
Hupmobile Eight-Cylinder Engine, 3 by 4¾ In.



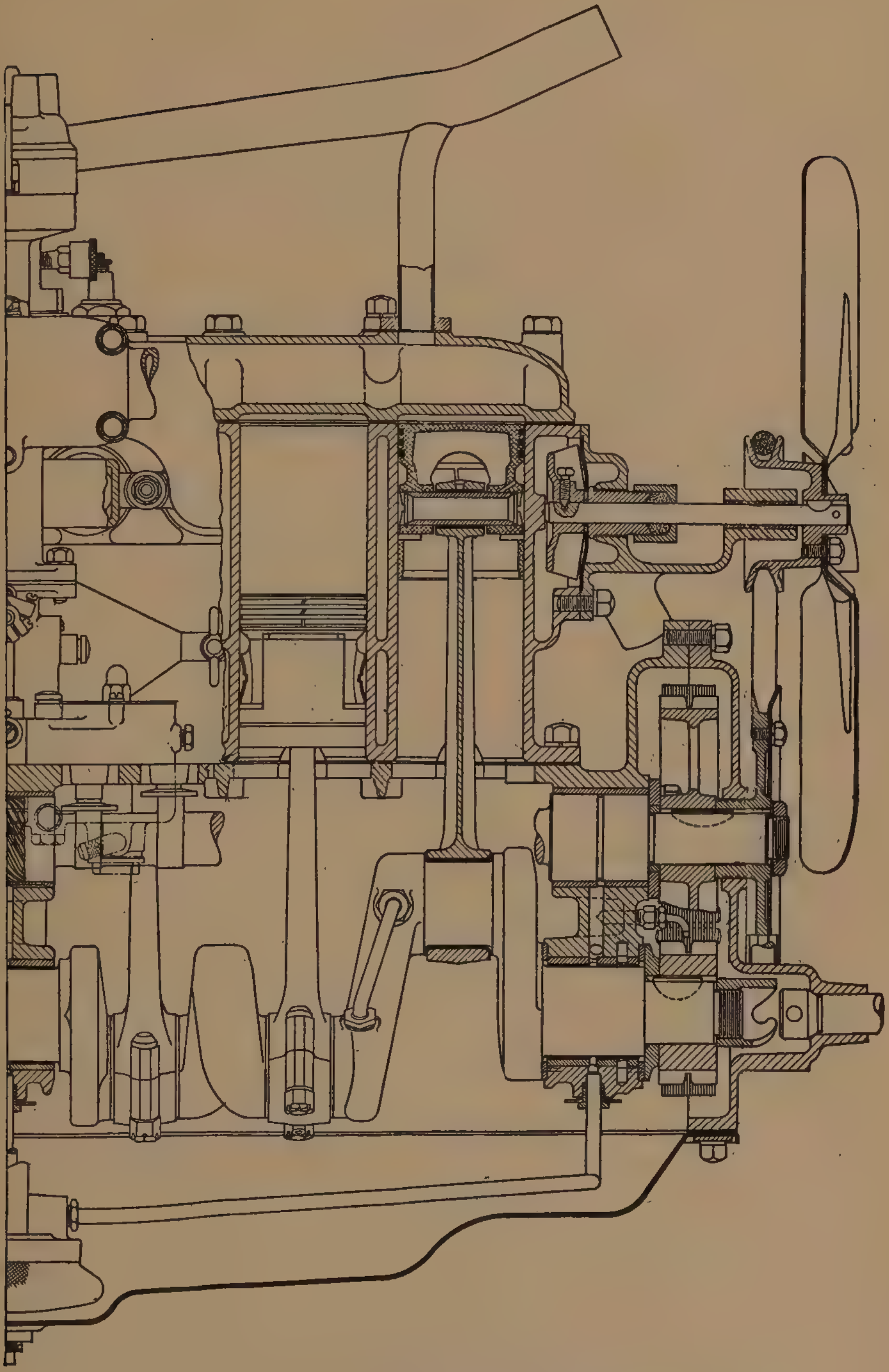
Hupmobile Eight-Cylinder Engine, 3 by 4¾ In.



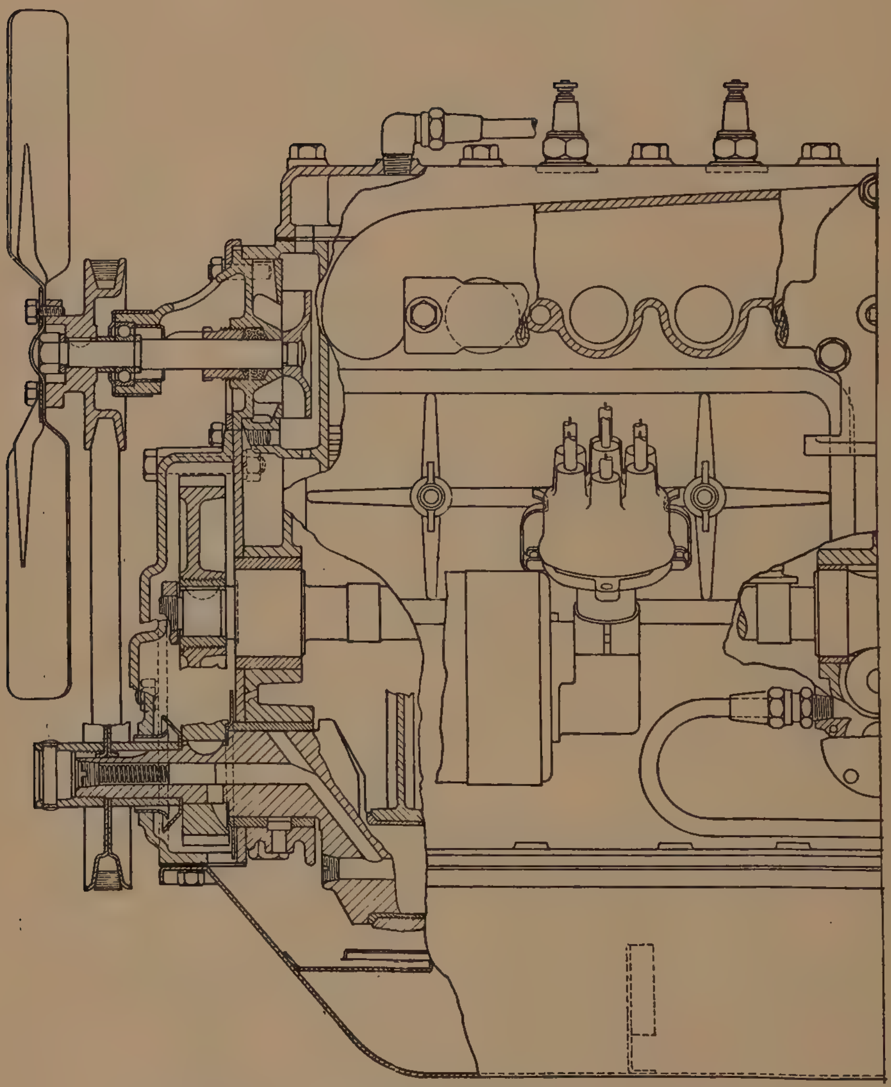
Oakland Six-Cylinder Engine, 2⅞ by 4¾ In.



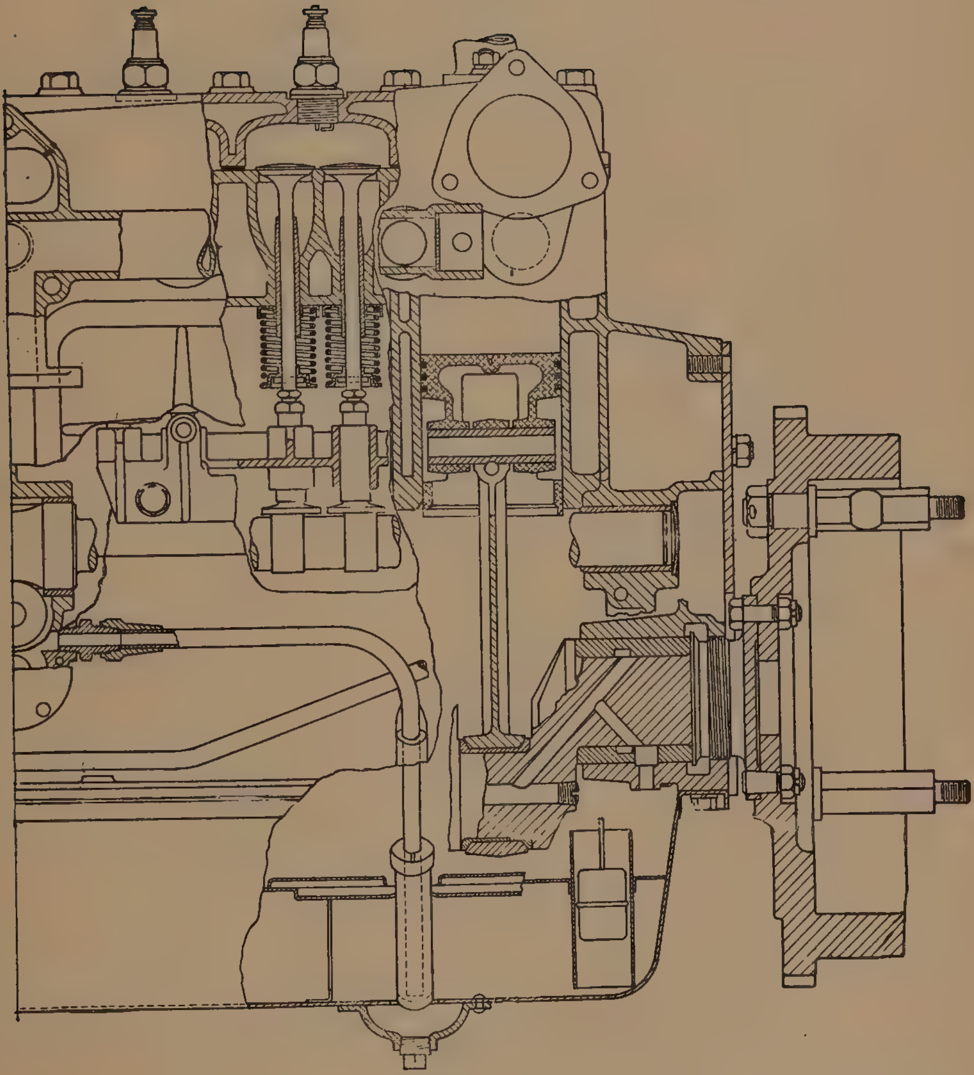
Oakland Six-Cylinder Engine, $2\frac{7}{8}$ by $4\frac{3}{4}$ In.



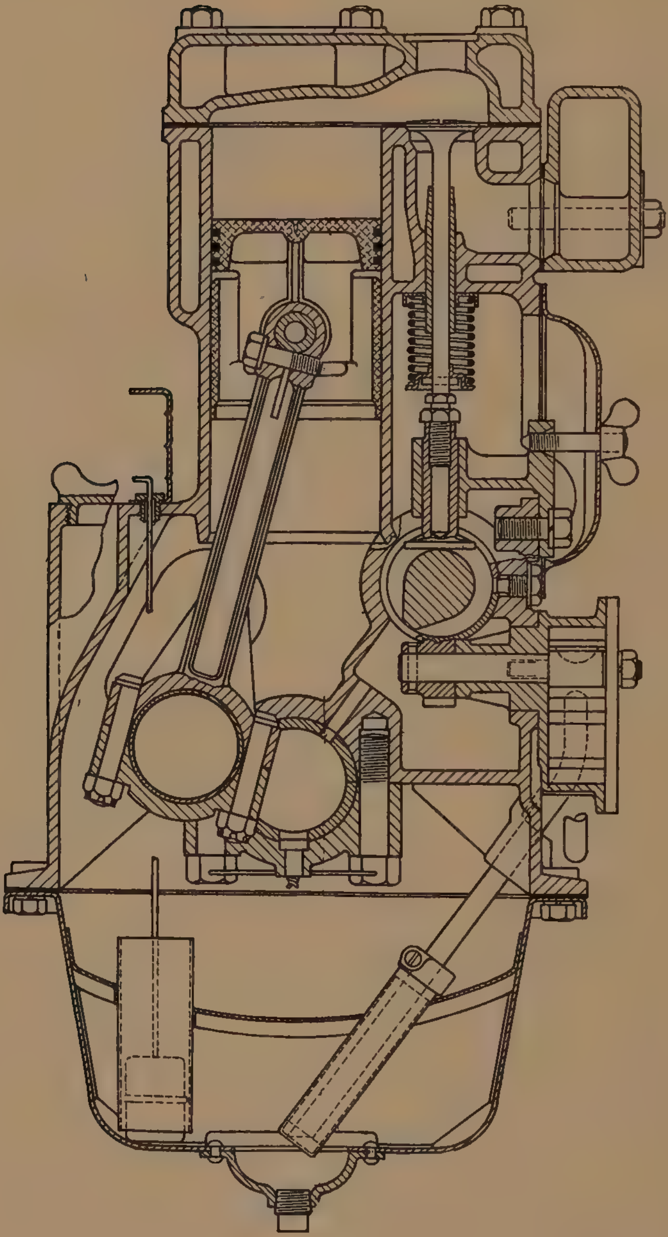
Oakland Six-Cylinder Engine, 2⅞ by 4¾ In.



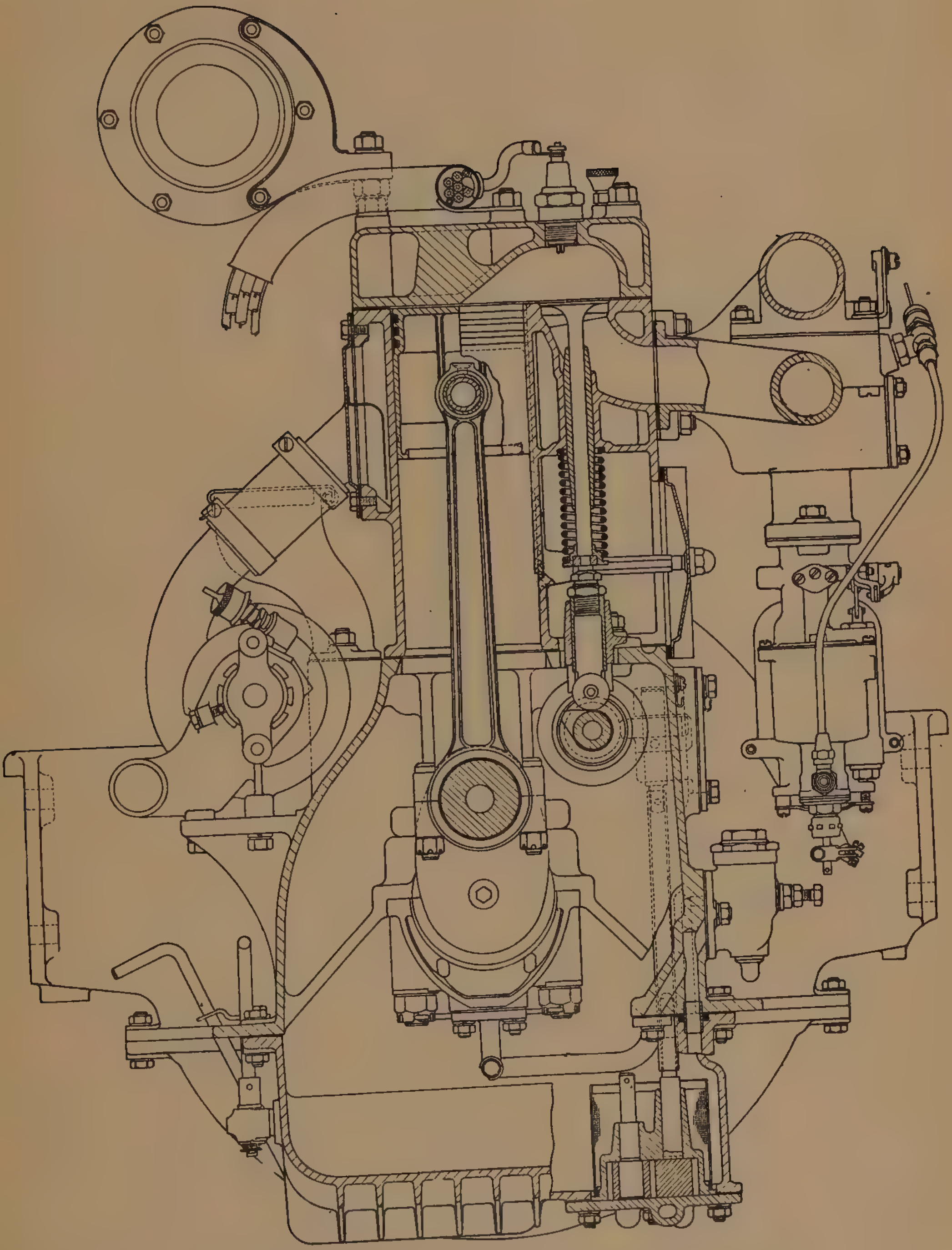
Overland Six-Cylinder Engine, 3 by 4 In.



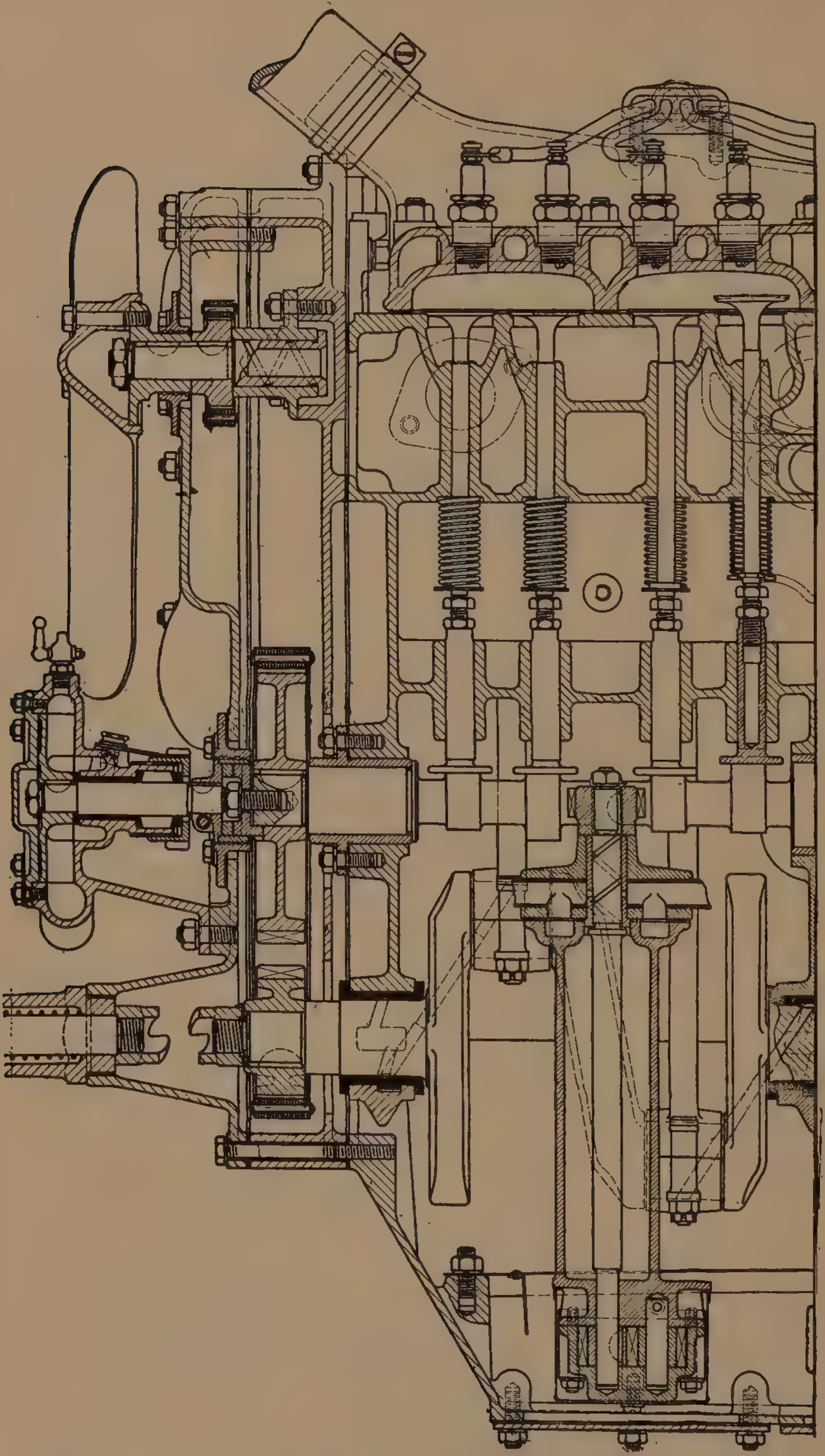
Overland Six-Cylinder Engine, 3 by 4 In.



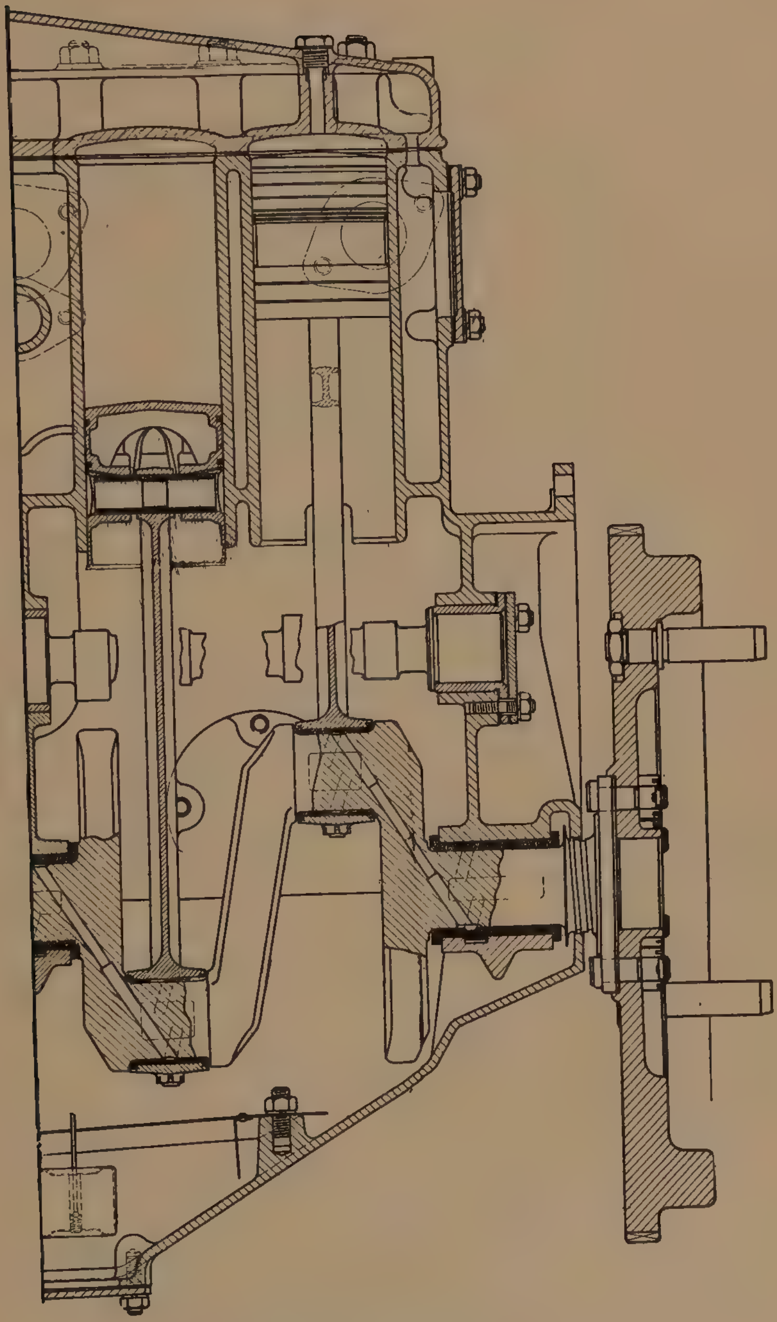
Overland Six-Cylinder Engine, 3 by 4 In.



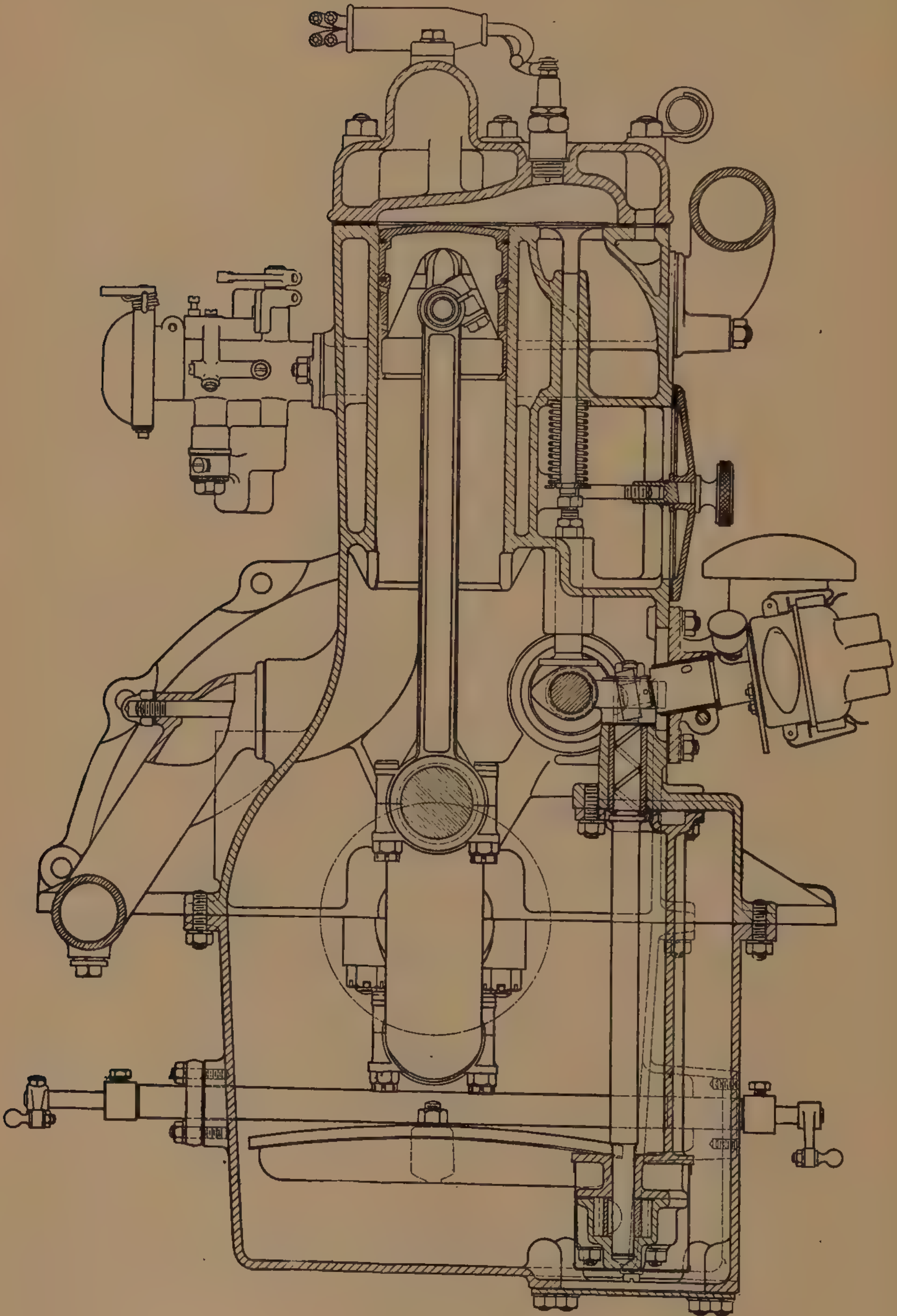
PIERCE ARROW SIX-CYLINDER ENGINE, 3½ BY 5 IN.



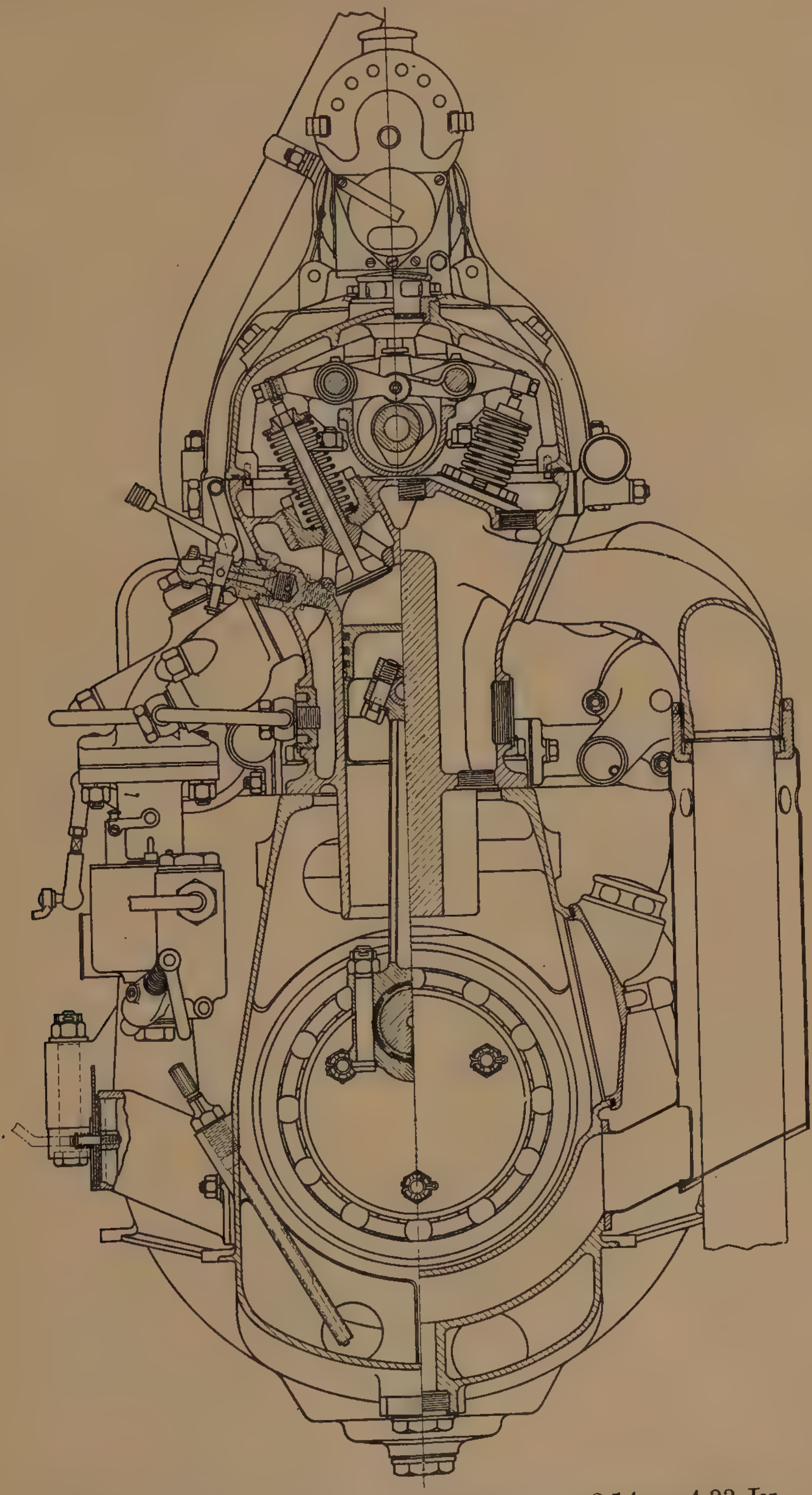
BEAN (BRITISH) FOUR-CYLINDER ENGINE, 2.9 BY 5.3 IN.



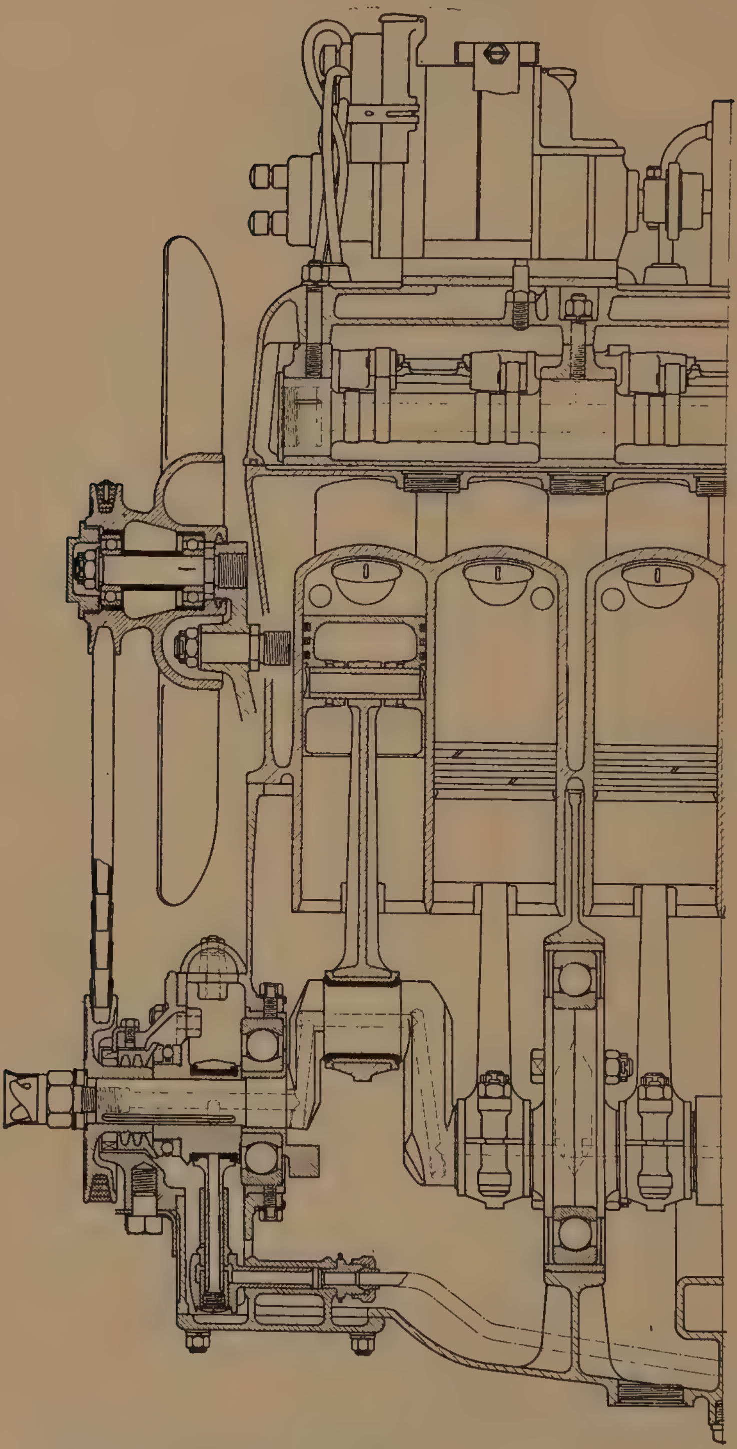
Bean Four-Cylinder Engine, 2.9 by 5.3 In.



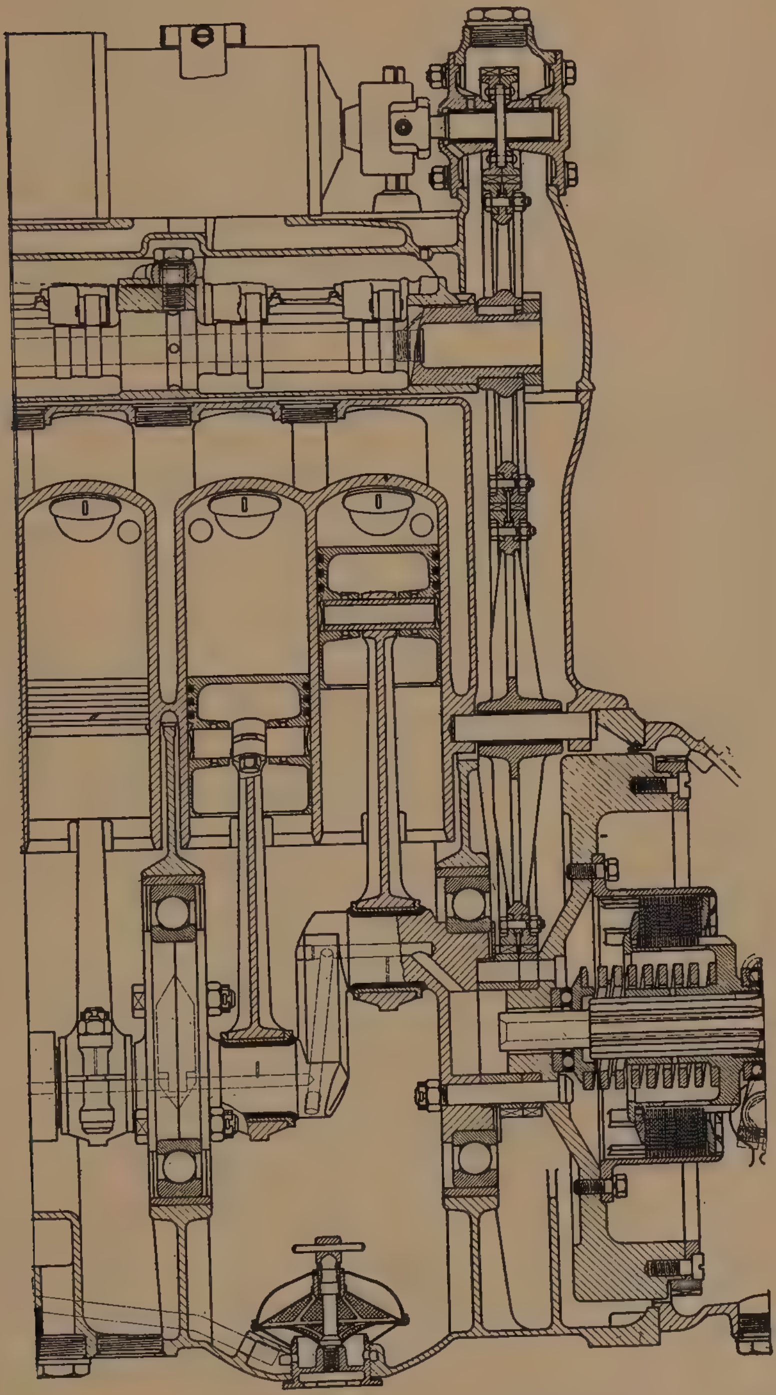
BEAN FOUR-CYLINDER ENGINE, 2.9 BY 5.3 IN.



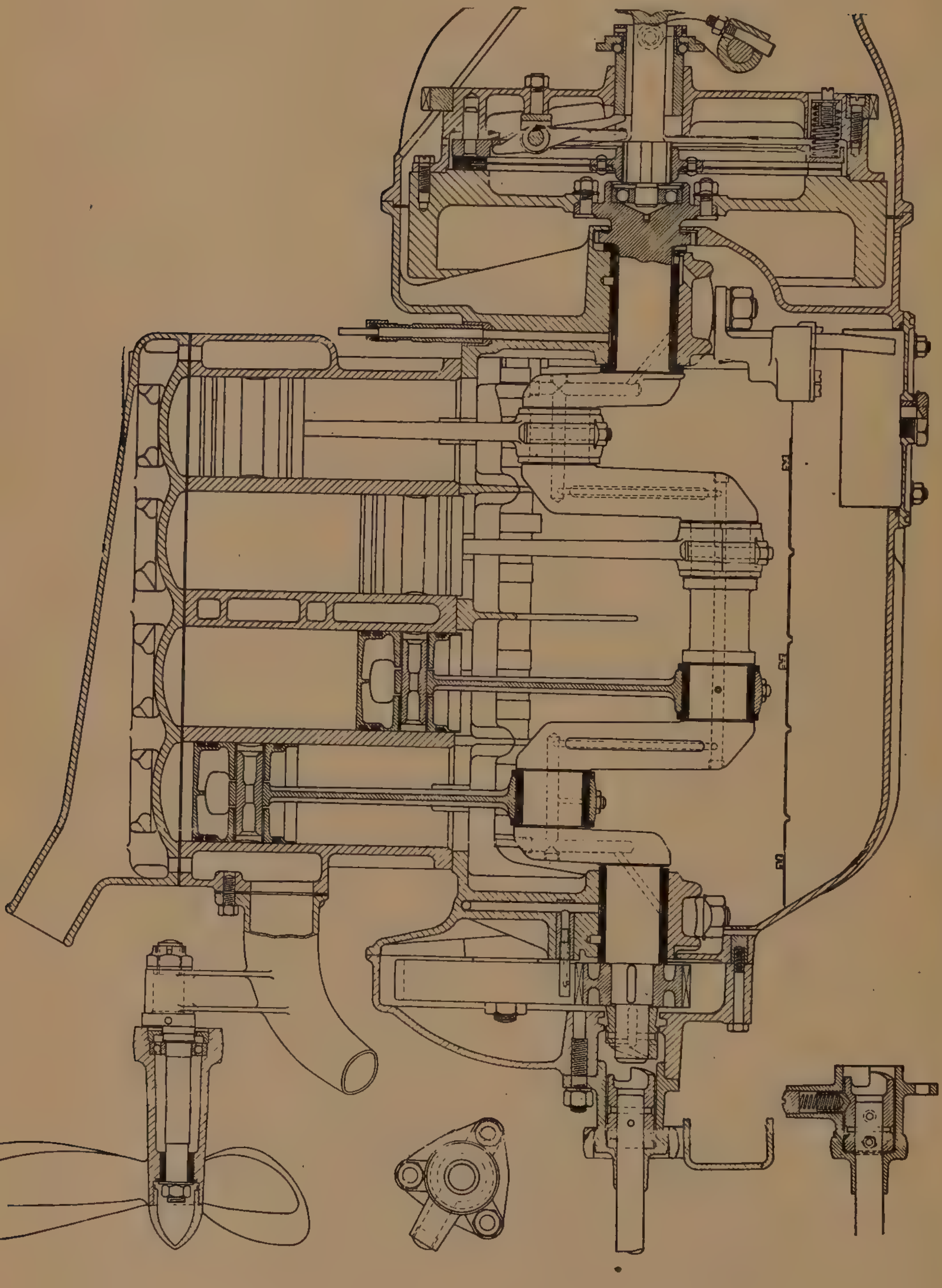
STEYR (AUSTRIAN) SIX-CYLINDER ENGINE, 3.14 BY 4.33 IN.



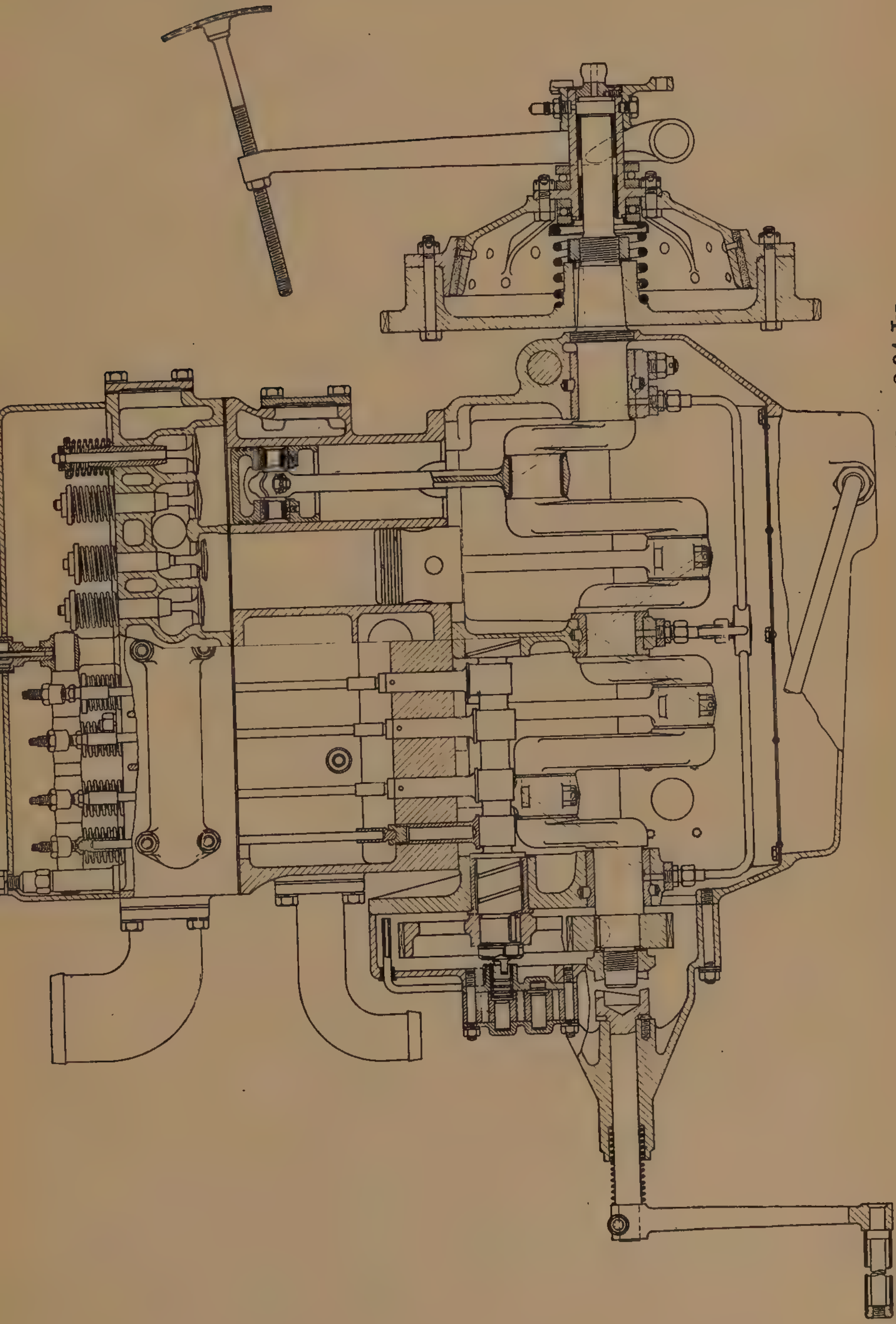
Steyr Six-Cylinder Engine, 3.14 by 4.33 In.



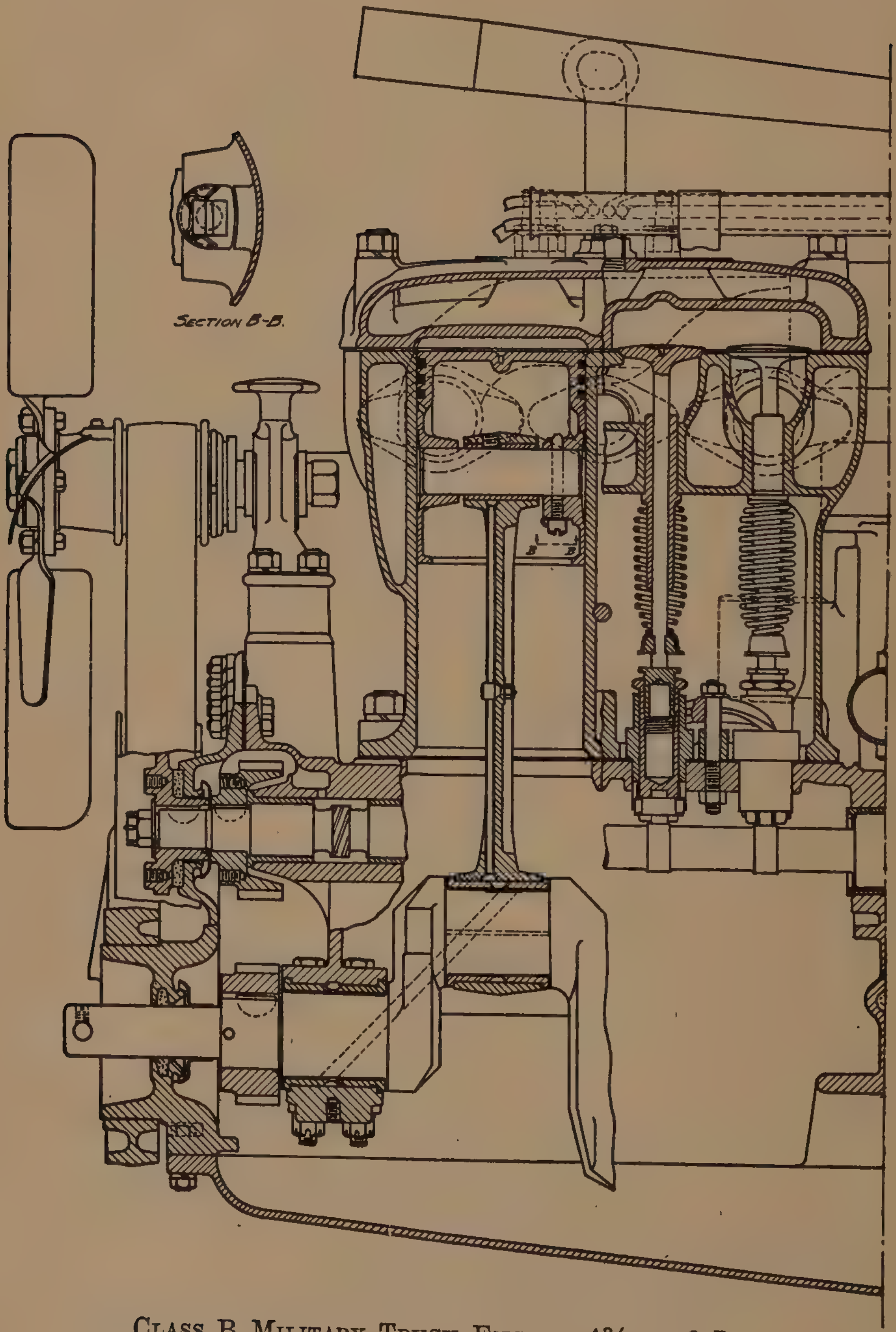
Steyr Six-Cylinder Engine, 3.14 by 4.33 In.



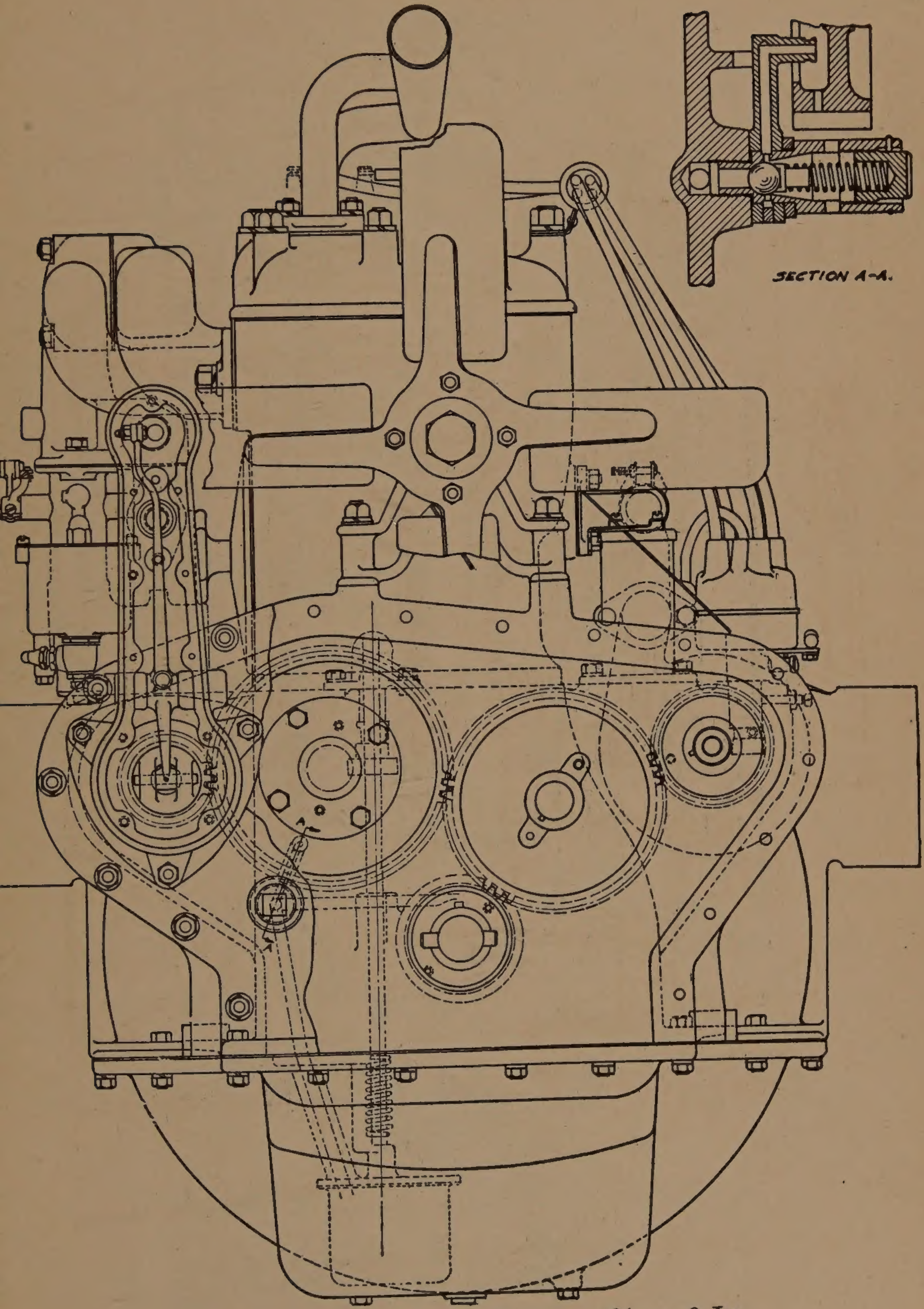
CITROEN (FRENCH) FOUR-CYLINDER ENGINE, 2.67 BY 4.33 IN.



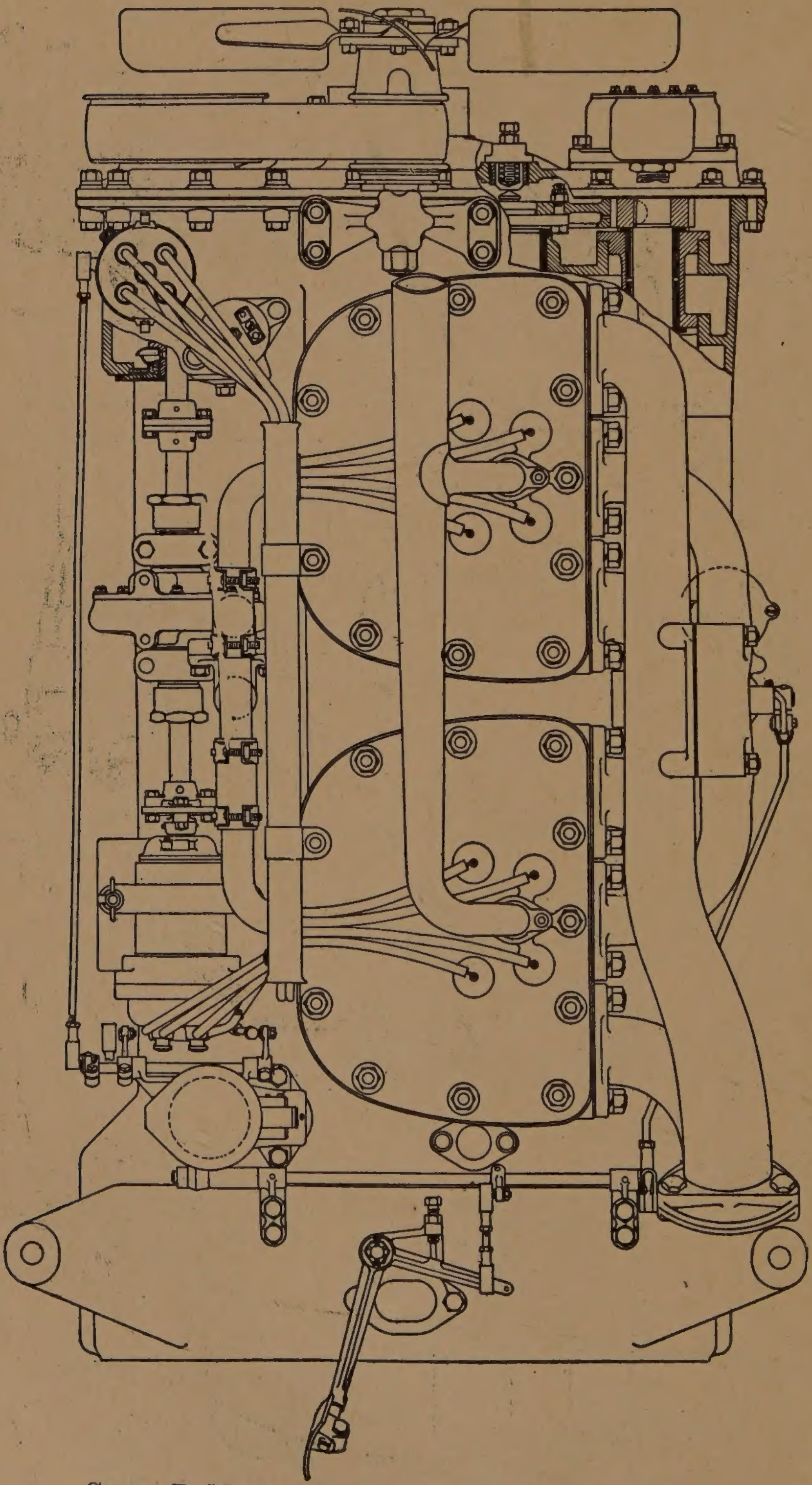
Gwynne (British) Four-Cylinder Engine, 2.16 by 3.94 In.



Class B Military Truck Engine, 4¾ by 6 In.



Class B Military Truck Engine, 4¾ by 6 In.



Class B Military Truck Engine, 4¾ by 6 In.

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